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(See editorial in this issue on The Mechanical Engineer and Our Country's Economic Progress)

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Wm. M. Frame

A. B. Kinzel

G. B. Karelitz

Charles Boehnlein

MECHANICAL ENGINEERING

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Has Taylorism Survived?

By DEXTER S. KIMBALL,[1] ITHACA, N. Y.

IN A PAPER by Lucien A. Legros, President of the British Section of the Société des Ingénieurs de France, entitled Economy of Human Effort in Relation to Industrial Fatigue, the following statement appears:

> It is a matter for regret that up to the present time no critical inquiry has been made into the extent of permanence in American industry of the work of Taylor, Gilbreth, and their followers. The work of these gifted pioneers was mainly directed to achieving definite objects, but it is difficult to ascertain from their writings whether the conditions were sufficiently stable to give permanence to the results, or how far they were applicable for conditions that varied.

In its discussion D. R. Wilson, of the Industrial Fatigue Research Board, comments as follows:

> Finally, Mr. Legros at the end of his paper alludes to the desirability of instituting some inquiry as to the extent to which the principles of Taylor have been applied in the United States. In theory I agree with him, but I am strongly disposed to think that the investigator would draw a blank. When I was in America, in 1924, I made many inquiries on this point, and found that from every quarter I approached came the names of the same four or five firms, and in such of these as I visited, only a very rudimentary form of Taylorism had been adopted.

These statements have attracted the attention of the author because they show a total lack of knowledge of what really has occurred in American industry during the last twenty-five years, and of the general tendencies in all progressive industries in this country. There is little difficulty in correcting the idea expressed in the first statement without going outside the Transactions of the A.S.M.E. In 1912 a special committee of the Society reported at some length upon the status of modern industrial management and that report is to be found in Vol. 34. Again in 1922 L. P. Alford presented the results of an exhaustive study of the progress that had been made in applying the teachings of Taylor and his associates and followers in a classic study entitled Ten Years' Progress in Management. This is to be found in Vol. 44. It is true that this study does not deal minutely with the question of fatigue, which is the subject of Mr. Legros' paper. But the question of fatigue itself is intimately connected with and dependent upon the other phases of management which Mr. Alford has so ably reported upon. Fatigue has been the subject of much study in American industry, but the discussions of this topic are quite widely scattered throughout the literature of management.

The second quotation is interesting from another angle, for it indicates a lack of knowledge of one of the most common phenomena in the history of the race and one which Kipling must have had in mind when he wrote—

> When 'Omer smote 'is bloomin' lyre
> He'd 'eard men sing by land an' sea,
> An' what 'e thought 'e might require,
> 'E went and took—the same as me.

In these lines Kipling expressed a great truth that has applied to every great reform movement that man has ever inaugurated. Every reformer is to a large degree the product and result of the times in which he lives as much as he is the product of the driving power within him. All great reforms and changes gather first as a nebulous cloud, the result of the work of many men, often working independently of each other and producing developments that, at the time, may not appear to be related to each other. Then comes some outstanding figure who, selecting a number of these apparently unrelated results, welds them into a new philosophy, adding perhaps some great contribution of his own, or possibly making no original contribution whatever. And because he gives definite shape and form to what had been previously only a *tendency*, he obtains a hearing and men flock to his support. As time goes on, however, it is found that the specific form into which he has molded the new movement is not a universal panacea and is not applicable to all individual cases, even though the elements of which his formula or organization is composed are all basic truths. Quite naturally, then, those who are interested select from these truths or principles those that are immediately applicable to their problem and develop a somewhat different interpretation of the original formula or panacea. The history of all great religious movements and reforms illustrates this process admirably.

All great reforms and changes gather first as a nebulous cloud, the result of the work of many men, often working independently of each other and producing developments that, at the time, may not appear to be related to each other. Then comes some outstanding figure who, selecting a number of these apparently unrelated results, welds them into a new philosophy, adding perhaps some great contribution of his own, or possibly making no original contribution whatever. And because he gives definite shape and form to what had been previously only a tendency, he obtains a hearing and men flock to his support. As time goes on, however, it is found that the specific form into which he has molded the new movement is not a universal panacea and is not applicable to all individual cases, even though the elements of which his formula or organization is composed are all basic truths. Quite naturally, then, those who are interested select from these truths or principles those that are immediately applicable to their problem and develop a somewhat different interpretation of the original formula or panacea. The history of all great religious movements and reforms illustrates this process admirably.

An excellent illustration is found in the work of Robert Owen, the father of welfare work, so called. Owen organized at New Lanark near Glasgow the first model industrial village, and there coördinated practically every instrumentality that has ever been used since in this work. But there are no model industrial villages exactly like Owen's to be found anywhere today. However, no one informed concerning this movement would say that Owen's work had perished. Some of his instrumentalities no longer apply because of changed industrial conditions and legal enactments; they have outlived their usefulness. But combinations of some of his methods in endless variety are to be found all over England and the United States. And more important still, the *spirit* of his reforms has remained a vital factor and an abiding influence in modern industry.

So it is with the work of Fred Taylor. Few, if any, enterprises will be found today that are organized exactly after the model set up by him at Bethlehem. But combinations of his methods in endless variety are to be found all over this country. It should be remembered that Taylor did not claim that he had invented a number of new management mechanisms. Most of the elements of his system such as cost finding, inspection, routing, dispatching, etc. were already in use in isolated and somewhat crude form. He simply took these mechanisms, refined them, added a few new ideas of his own, molded them into a new philosophy of management, and illustrated this philosophy by a concrete example which produced

[1] Dean, College of Engineering, Cornell Univ. Past-President A.S.M.E.

the practical results desired. Without this concrete and successful example it is doubtful whether the movement would have readily gained headway against the inertia of old habits and inefficient practices, and Taylor in all probability would have been remembered only as another idealist.

But when others tried to use these methods exactly as Taylor had enunciated them it was found that they did not fit the circumstances, and after some rather disastrous experiences in trying to make the circumstances conform to the classic example at Bethlehem, the thoughtful manager began to select those mechanisms that fitted his needs and ceased to worry because he could not use all the mechanisms employed by Taylor; or in the words of the poet:

> An' what 'e thought 'e might require,
> 'E went and took the same as me.

To say, therefore, that an investigator of the Taylor methods in the United States "would draw a blank," is to indicate gross ignorance of just what has occurred. Mr. Taylor's great paper Shop Management still remains the basic statement of a new industrial philosophy. Many interpretations, however, have been made of this philosophy, just as many and diverse interpretations have invariably been made of the work of all great reformers, religious and political. But any one well informed concerning Taylor's pioneer work will find no difficulty in identifying some of his methods in almost any up-to-date manufacturing plant in this country. And in many of them he will find some of Taylor's mechanisms of management almost unchanged from the original. Perhaps the most common elements to be found are those of rate setting on the basis of accurate time study, and the planning and dispatching of operations on the basis of a prearranged time schedule.

But more important and underlying any and all of these mechanisms of management that may be identified is the spirit of the man himself as the prophet of more efficient methods. Whether any or all of his specific methods survive, the spirit of inquiry that he set into motion concerning industrial methods, his frank skepticism of the efficiency and desirability of methods and processes, even though they bore the imprint of hoary age and the stamp of ancient precedent, will ever remain one of the great contributions to the industrial arts. At no time has the influence of Fred Taylor been so great or his memory so secure as at this moment when the factories of the United States are pouring forth a stream of industrial wealth such as mankind has never witnessed. His share in this great development is great, indeed, and difficult to estimate.

The Fuel-Oil Situation in the Central States

IN A PAPER presented before the Second Mid-West Power Conference, held in Chicago, Ill., February 15 to 18, 1927, Campbell Osborn discussed the present and probable future fuel-oil supply in the central states, also the uses of fuel oil in this area.

He pointed out that in former years nearly all the gas oil and the residual oil obtained in this country as by-products in the fractional distillation of crude petroleum for gasoline was consumed for fuel. Later the demand for gasoline increased more rapidly than the production of crude oil, the price of gasoline rose, wide differentials prevailed between fuel oil and gasoline, and it became feasible and profitable to crack fuel oil for its gasoline content. Now one-third of the nation's gasoline requirements is obtained from fuel oil. This profound change has brought about a lower ultimate yield of marketed fuel per barrel of petroleum charge to stills, a new type of fuel known as "cracked fuel oil," and a higher price.

In recent years many predictions have been made that the peak of petroleum production has passed and that the time is close at hand when fuel oil will be so high that it can no longer compete with coal. The author's belief was that the peak of production might ultimately be determined by the peak demand, and that it would come many years hence. As the best means of conservation, he favored more efficient production and use. Oil being a superior fuel, he urged taking advantage of its superior characteristics.

Regarding price competition of fuel oil and gasoline, a petroleum expert has made the prediction that equal competition will be realized when a total yield of 42.5 per cent of gasoline by combined cracking, skimming, and blending methods has been reached. Mr. Osborn did not agree with this figure, his opinion being that it would be nearer 46.4 per cent, which would mean that instead of reaching the limit in two years, as would be the case with the former figure, it would not be reached until 1930. Explaining, he said that the total United States supply of fuel oil produced in 1925 was 460,000,000 barrels. If this entire amount were subjected to cracking under present practice there would be a loss in the process of extracting the gasoline amounting to 8.7 per cent, or roughly 40,000,000 barrels, which would consist of gas and coke. The quantity of fuel oil consumed in the United States in 1925 for the superior uses, exclusive of cracked gasoline, was 226,000,000 barrels. If this amount of fuel oil, plus the loss referred to, be deducted from the total supply, there would be a remainder of 192,000,000 barrels of gasoline. This, plus the yields of skimmed gasoline actually obtained in 1925, he said, justified the percentage figure above cited, which, he also explained, was not what he considered the ultimate end of cracking.

Until recent years, he explained, the importance of cracking had not been fully appreciated in the central states. When gasoline was higher and crude oil abundant profits could be made by skimming, and the smaller plants hesitated to make the large investment necessary for cracking. Now, however, the situation has changed and cracking is more general. In consequence, the percentage of increase in total gasoline yield has been at the rate of about 3 per cent per year during the past five years, as compared with 2 per cent for the rest of the United States, and the proportion of the total gasoline obtained by cracking at the close of the year was 39 per cent, as compared with 33 per cent for the entire country. This would indicate that the yield of marketable fuel oil per barrel of crude charged to stills in the central states might not be reduced by as much during the next several years as it would be in some other areas, particularly California.

Summarizing, the outstanding conclusions reached by the author are as follows:

There probably will be no drastic change in the supply of fuel oil for efficient users in the United States during the next several years. Increased runs to stills are expected to offset decreased yields per barrel of crude. It may be necessary to improve burners so that they will satisfactorily and efficiently burn cracked fuel oil, and to install more Diesel engines. It is uneconomical to burn fuel oil directly in competition with coal on a purely B.t.u. basis, and the use for this purpose will be reduced.

Cracking will further increase, but not as rapidly as in some other parts of the country. It is felt that most of the fuel oil currently produced in the Mid-Continent is being subjected to cracking.

The consumption of oil in Diesel engines will increase, despite a rise in price. The Diesel engine is one of the most economic sources of power from oil.

During the recent past the railroads have met their increased requirements for oil by increasing efficiency. It is doubtful whether the volume they use will again increase except under the stimulus of overproduction. The Diesel-electric locomotive now in the experimental stage may some day become a substantial consumer of fuel oil.

The artificial gas plants can now probably compete equally with the gasoline buyers for their requirements.

The use of fuel oil for heating homes and office buildings will increase, because the commodity is largely bought on the basis of convenience, for which the public is willing to pay. This expansion will be slow at first and will depend upon the education of the consumer, further improvements and standardization of burners, and better service after installation.

The use of fuel oil to generate steam for heat and power in inland industries other than railroads is declining.

With a stable crude supply, prices of fuel oil are expected to further slowly increase until cracking has proceeded to the point in the United States where the total yield of gasoline exceeds 45 per cent.

Morale as a Factor in Time-Study Technique

As Illustrated by a Recent Investigation of the Production Standards Used in the Garment Industry in Cleveland, Ohio

By MORRIS LLEWELLYN COOKE,[1] PHILADELPHIA, PA.

PERHAPS it should be recalled right at the start that effective time study has never been conducted outside the worker's knowledge and participation, and that the coöperation of the particular individual whose work is being analyzed is requisite to satisfactory results. Now, however, it is becoming evident that group reactions vitally condition the effectiveness of time study. More and more we recognize time study as one factor in a total situation and not as a thing apart. Morale—the state of mind of the group—has become precedent to the proper practice of time study, and to any reasonably accurate determination of production standards for the individual.

Because of time limitations the author will seek to confine the discussion to that phase of time study which has for its purpose the determination of the quantity of a given kind of work which can and should be performed by a given worker in a specified unit of time. These measures of the output of the individual worker are referred to in the Cleveland Agreement about to be described as "production standards." This treatment will be to the exclusion of such aspects of time study as (1) the standardization of process, which is assumed to be a wholly necessary preliminary, and (2) the inducements, financial and non-financial, designed to insure the performance of given tasks. No special attempt will be made to develop the arguments which seem to favor time study as an essential factor in the industry of the future. In this connection, however, it is important to have in mind that time study *has* a function to perform in industry quite distinct from the determination of the skill or productivity of the individual worker. As a means of detecting waste motions and wasted effort generally and as a guide in determining the one best way, process engineering will increasingly depend upon time study.

During the war many of the garment[2] manufacturers of Cleveland were engaged in Government-uniform contracts. Largely through the good offices of the then Secretary of War the employers of the market and their organized employees[3] were led to enter into a coöperative agreement in order "to preserve peace in the industry and to further their mutual interests in the common enterprise." The preamble to this agreement reads:

> In view of their primary responsibility to the consuming public, workers and owners are jointly and separately responsible for the cost and quality of the service rendered. It is agreed that coöperation and mutual helpfulness are the basis of right and progressive industrial relations, and that intimidation and coercion have no proper place in American industry.

The agreement further holds:

> That it is due to the consuming public whose patronage supports the industry as well as to the very existence of the industry in Cleveland itself, that all activities, decisions, and arrangements growing out of the Agreement shall be based on the principles of true efficiency and the necessity for the lowest unit cost of production possible under the wage scale as determined by the Referees.

Space will not permit even a summary of the various points covered by this Agreement, which during the eight years of its existence has been wholly successful in keeping peace in what has usually been considered a turbulent industry. The administration of the Agreement is, in the first instance, in the hands of a committee of two,[4] a representative of the employers and a representative of the workers. Disagreements, which fortunately have been unexpectedly infrequent, are referred to an impartial chairman,[5] a person wholly outside the industry and chosen by both sides. A Board of Referees also mutually chosen and recruited from without the industry, fixes the wage scale annually. It may be revised at six months' periods on the request of either side, with a change in the cost of living affording the only basis for a change either up or down.

Agreement Between Cleveland Garment Workers and Employers a Wide Departure from Ordinary Industrial Procedures

Two outstanding features of the agreement are (1) the concession of a guarantee of 40 weeks of work on the part of the employers, and (2) the acceptance of measured production for the individual employee on the part of the union through "a weekly minimum guaranteed wage for each worker and an additional wage depending upon his or her production measured by standards based upon time studies."

It is provided that "the unit of measurement shall be the production of a worker of average skill working at normal speed for a week of 44 hours."

There is a committee on standards recruited from the workers, named by the union in every department of a shop. This committee approves or disapproves the standards submitted to it. In case the standards submitted are rejected either by the committee of the union or by the employer, the matter is taken up by the representative of the union and the representative of the association. In case of a disagreement it is provided that these two shall select "a third party who shall be a person skilled in the trade." The decision of this third party is final.

Thus it was early recognized that disputes about standards (i.e., the determination of the quantity of work that is to be performed in a given time by an individual employee) involve questions highly technical in character which do not lend themselves readily to equity procedures. In stipulating that such disputes were to be settled "by one skilled in the trade," it is probable that the intent was to discourage such disputes by practically throwing them outside the Agreement. At any rate, during the first five or six years of the Agreement such disputes as arose about standards were ironed out in the shop. Once in 1924 and again in 1925 the author, notwithstanding his lack of "skill in the trade," was asked to hear and settle disputes of this character.

The following quotations from the decisions rendered in these cases bear directly on our subject:

> 1 Our procedures wisely afford the employees, through an appropriate committee, an untrammeled opportunity to comment upon and even to criticize each and every standard set up. This relates to any given standard as a whole as well as to any of the elementary units or parts of which it may be composed. Any restriction of this opportunity for critical comment on time-study data would not only be entirely contrary to the spirit of the Agreement, but would not make for that basis of understanding, as between the technicians and the workers, which is essential to progress.
>
> However, the undiscriminating refusal of an entire line[6] is not at all of a piece with either the Agreement or a scientific approach to industry. Such a sweeping rejection of time-study work—presumably carried on in good faith—if made with a full understanding of its implications, would be the equivalent of rejecting standards—perhaps not the most essential idea underlying our Agreement—but certainly one which, if abandoned, would require a radical revamping of the understanding under which the market has been satisfactorily operating for over five years.
>
> 2 The employees have rejected the standards set for virtually the entire line largely on the ground that the earnings under them showed a decided falling off as compared with previous seasons. Assuming reasonable performance on the part of the manufacturer, and especially on the part of the time-study man who determines the standards, any such wholesale

[1] Consulting Management Engineer. Mem. A.S.M.E.

[2] Technically women's cloaks, suits, and dresses are known as "garments," while men's suits and overcoats are known as "clothing."

[3] Members of the International Ladies' Garment Workers Union.

[4] Messrs. Abe Katovsky, Business Agent of the Cleveland Joint Board of the International Ladies' Garment Workers Union and Fred C. Butler, Manager of the Cleveland Garment Manufacturers Association.

Paper presented at a joint meeting of the Metropolitan Section of the A.S.M.E. and the Metropolitan Section of the Taylor Society, New York, March 9, 1927.

[5] Dr. Jacob H. Hollander, Professor of Economics, Johns Hopkins University, Baltimore, Md.

[6] This term is used as including all the season's models, in this case numbering 70 to 80 individual garments.

rejections would be tantamount to a violation of the spirit of the Agreement. Styles do not change so completely from season to season as to warrant wholesale rejections. Standards mutually acceptable one season are, other things being equal, not subject to question the following season. In other words, assuming a proper analysis (through time study) of a given piece of work and of the time taken on its individual elements as well as the orderly maintenance of clear records of previous observations and standard determinations, such wholesale rejections are entirely out of order. Such criticisms of standards as are made must be reasonably discriminating and capable of detailed statement and analysis.

Criticisms once raised must likewise be met, not by generalities but by reference to the data which are in such shape as to be reasonably convincing even to the layman. After all, standards (shop standards, market standards, and some day inter-market standards) will only come into general use as they are understood by and command the confidence of those who work under them as well as those who make the determinations. It is a part of the duty of the time-study man to interpret his work to those most intimately affected by it. Access to time-study records should always be reasonably easy. The records themselves should be maintained in such a way as to make them understandable and broadly informative.

As a direct result of the hearing of the last of these two cases Mr. Francis Goodell, an industrial engineer and a member of the Taylor Society, was employed at the joint expense of the Union and the Manufacturers Association to make a study of the whole standards situation. A copy of his report of this inquiry[7] with comments thereon by the various parties at interest has been filed in the Engineering Societies Library. As this report has the distinction of carrying the first discussion of the technique of time study as viewed by organized workers, this concluding paragraph of the Union's letter of comment is not without significance:

In conclusion we beg to advise that our proposals for the further improvement of the standards system are submitted in a constructive spirit. We

Fig. 1 Making Time Studies on "Gathering"
(The operative is a member of the Union and the observer is jointly paid by the union and the manufacturer.)

agree with the thoughts expressed by Mr. Goodell in his report and by Mr. Cooke in his letters of transmittal concerning the value and soundness of the system and the progress made toward its realization. We stand ready and willing to do what is in our power as a union of constructively minded, skilled, and capable garment workers to make our contribution to the further improvement of the standards system.

It would be a mistake to overemphasize the importance of this Cleveland departure from usual industrial procedures. The number of workers involved is small and the Cleveland garment market is not by any means a dominant factor in the garment industry. The resources available are necessarily limited. In appraising the results we have to recognize that on both sides there always have been, and still are, serious misgivings as to the operation of the plan. The Cleveland manufacturers, by virtue of their geographical location, have had difficulty in securing a proportionate share of the trade as against the more favored Chicago and New York markets. But that in spite of handicaps these novel industrial procedures—involving severe disciplines self-imposed upon employee and employer alike—have survived affords strong testimony in their behalf. There is certainly no thought in any one's mind as to their abandonment.

In Mr. Goodell's report both time study and standards are discussed with a range much wider than is possible in the present paper. Further references to it and the Cleveland agreement will therefore be made only as they contribute directly to the development of the author's theme.

The fact that the Cleveland garment workers constitute a section of one of the more important needle-trade organizations, which

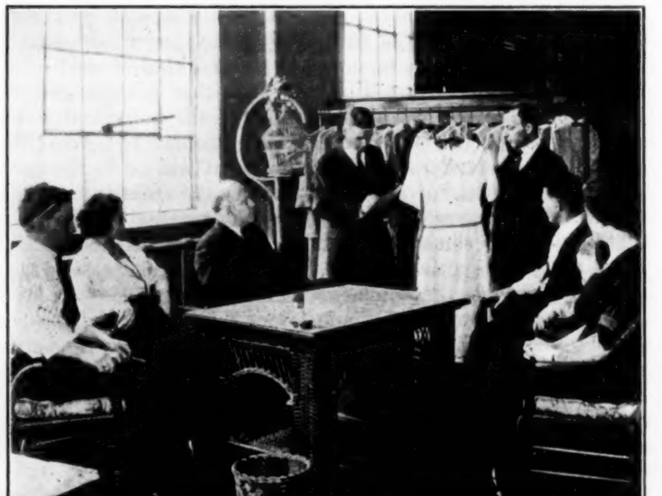

Fig. 2 Production Standards Under Review by Department Committee
(Present: the employer, the time-study man, and representatives of the workers.)

in turn is affiliated with the American Federation of Labor, is quite incidental. The discussion applied to so-called company unions or to any situation where some degree of group solidarity has been established or where group reactions are in any way recognized or valued.

Indeed, such a recognition of the groupings of the workers in industry is forced on us by the growing integration of the industrial process. We have come to realize that the production mechanism cannot break or even weaken at one point without endangering the total effort. The effects of discord and lack of understanding can no longer be wholly localized in view of the interdependence of all the agencies of production. Because, for instance, there must be maintained a nice balance as between sales, finance, and production, the enthusiasm of the sales force—even the attitude of the investing public—may strongly influence the men and women at work on the product. So we come to reckon with the collective mind as well as with the reactions of individuals. Further progress in establishing the art and science of time study will become more and more dependent upon morale interpreted as the state of mind not of one group but as that totality of attitude resulting from those various groupings and regroupings through which we members of the human family are related to any given undertaking.

As first practiced, time study was concerned almost wholly with the individual. Taylor recommended in the early days that an operator while being studied should, as far as possible, be isolated, and further suggested that while being observed he be given some special compensation to insure active participation in the work. As the mechanisms of time study became more familiar in the shop and their purpose better understood these measures were neither

[7] A report on the Production Standards Situation in the Ladies' Garment Industry of Cleveland with comments by officers of the Employers Association and of the Union, August, 1925.

necessary nor desirable. So that today, except for a negligible amount of laboratory research, time studies of individuals are made at the regular work places and in full view not only of the operator but of his associates. This development is especially suggestive in view of the larger number of persons performing like operations in present-day manufacturing plants.

Defects in Current Time-Study Practice as Affecting Group Relationships

The most serious defect in our current time-study practice—as it affects and is affected by these group relationships—lies in the almost complete inability to take advantage of the softening influences of publicity. Notwithstanding the fact that our first time studies were made over 40 years ago, there is today no generally recognized code under which observations are made or recorded. The consequence is that with negligible exceptions time-study data are individual to the observer, and except in the hands of the person who made the observations cannot be used.[8] The systemization of our data is generally too informal to permit of its being grasped by the layman. For every variety of work it is possible to have at least a reasonably solid substructure of time-study data relating particularly to "allowances" applicable to a given shop and to operations and sub-operations most frequently encountered and those least likely to change, which can be passed around somewhat as we do the multiplication tables. We are more tolerant of variation where some part of the structure is reasonably stable. In the author's use of the term "time study" he has in mind of course that painstaking effort to get at bottom facts which involves not only the division and subdivision of the work to be done into its elementary parts, but timing observations on each such part in such number as to practically eliminate guesswork. He does not have in mind those overall approximations and guesses which are current in industry under a dozen alluring names.

Curiously enough, the practitioners of time study have never banded themselves together, even as a group within a parent organization. Perhaps this is the reason for an almost complete lack of a commonly accepted technique. Certainly the fact that no such organization exists, reflects no credit on the art. It will not profit us, perhaps, to speculate on all the reasons as to why this numerous group of technicians have never sought to establish this quite common means for the exchange of views and the stabilization of practice. Certain it is that any such organization would necessarily blazon at the top of its program an emphatic prohibition against secrecy in every phase of time-study work. There can be no reserve in condemning those who may be using undercover methods in this field. Observing workers without their knowledge and consent and participation is wholly bad.

When such an organization as has been indicated comes into existence, one of its first tasks will be to prepare a very simple treatise on time study—in leaflet form—for the information of the men and women in the shop. In view of the small amount of information which the general public has as to objectives and methods of time study, we are assuming quite too much in asking that same public to accept its conclusions.

The literature of time study is replete with references to the difference in productive ability as between a worker designated as "first class" and one who is rated as not "first class"—this latter sometimes referred to as "an average worker." In the absence of more exhaustive research than has yet been given to this subject, one must tread lightly. But the author may be permitted to raise the question as to whether the difference between (1) what might be called "normal" men and (2) what are known as "sports" in biology may not be the real consideration.

Taylor says:[9]

> That there is a difference between the average and the first-class man is known to all employers, but that the first-class man can do in most cases from two to four times as much as is done by an average man is known to but few, and is fully realized only by those who have made a thorough and scientific study of the possibilities of men.
>
> The writer has found this enormous difference between the first-class and average man to exist in all of the trades and branches of labor which he has investigated, and these cover a large field as he, together with several of his friends, has been engaged with more than usual opportunities for thirty years past in carefully and systematically studying this subject.
>
> This difference in the output of first-class and average men is as little realized by the workmen as by their employers. The first-class men know that they can do more work than the average, but they have rarely made any careful study of the matter. And the writer has over and over again found them utterly incredulous when he informed them, after close observation and study, how much they were able to do. In fact, in most cases when first told that they are able to do two or three times as much as they have done they take it as a joke and will not believe that one is in earnest.
>
> It must be distinctly understood that in referring to the possibilities of a first-class man the writer does not mean what he can do when on a spurt or when he is over-exerting himself, but what a good man can keep up for a long term of years without injury to his health. It is a pace under which men become happier and thrive.

Taylor wrote this over twenty years ago (1903), before modern psychology had discovered that in all groupings any characteristics are spread over a frequency curve in such a way as to present an altogether new picture of variability. It is no longer scientific to classify on the "white or black—good or bad" basis, but on the basis of a recognized norm and a dispersion about it. Therefore in seeking operatives best fitted for a given class of work we shall probably be well advised to look neither for "speed boys" nor subnormals but for high-class normals, doubtless corresponding to what Taylor had in mind in using the term "first-class worker," "a steady, intelligent and conscientious worker, skilled in the trade, who produced good work and whose performance might under proper conditions and instructions reasonably be expected to be attained by any one physically, intellectually, and temperamentally suited to the work."[10] It is, generally speaking, possible of course to find highly exceptional men. But just how advantageous it is to look for them in industry is another matter. My own point of view is that of Col. H. K. Hathaway—a member of these Societies—quoted at length in my introduction to the Cleveland report. No one interested in the philosophy of time study can afford to miss this statement of Colonel Hathaway's—summarizing as it does a peculiarly ripe experience in this field. An all-star cast on special occasions may draw an audience. But such aggregations of talent are not usually continuing commercial successes. And even more important, they put a burden on the manager which even profits may not sustain in the long run. As with the stage, so it is likely to be in industry.

From the standpoint of the workers the statement of Colonel Hathaway's that the use of studies of "average" performance may actually result in cutting rates is especially interesting:

> For example [he says], let us consider that time studies are made as indicated by the report, on employees of widely varying degrees of skill and as a consequence average production standards are set. Theoretically under such an arrangement the various workers earn proportionately to their ability to produce. Gradually, however, as a result of improvement in management and all of the conditions involved as well as of self-development of the workers, the difference in ability between the various workers diminishes until they are all within reasonable limits equally competent. At the same time changes in product, method, etc., gradually come about and new studies are made. The base or minimum wage remains the same. Now, under the earlier state of affairs the earnings of the average worker would have been, let us say, 100, and that of the first-class worker say, 50 per cent higher, while if, as is not so improbable as it may sound, the gradual improvement mentioned resulted in the entire force becoming first-class workers, we would find them all earning the figure represented by 100 instead of 150. I am confident that with a reasonably constant force I could bring this about [i.e., the raising of the *average* level of skill and of performance to the level of the *first class*] in from two to four years. Of course there would be a few exceptional people who could exceed the standard as well as a very few who might never attain it, but in setting standards we are not in the first instance concerned with either of these.

From the standpoint of the group these conclusions if valid have an important bearing on the attitude to be taken toward time study. Naturally no group is going to be wildly enthusiastic about any system under which a considerable percentage of its membership is to be barred from further activity. But if, as appears to be the case, it is possible through standardization of process, providing the proper service for the worker, coaching, and otherwise, within a very short time to carry the great majority of a group of workers to the level of first-class performers and to the enjoyment of the higher earning power (and this without prejudice to their associates!), we can expect to see the great body of workers won

[8] There are of course notable exceptions—such as the Link Belt Co. and the Tabor Manufacturing Co., where the original work was so thoroughly done that 15–20 years later it is still in active every-day use.

[9] Shop Management, pp. 24–25.

[10] See Cleveland Report.

over as the Cleveland garment workers have been to the advantages of time study. Colonel Hathaway is undoubtedly right in pointing out the futility of time studies based on the performance of "average" men. It is only time-study data based on reasonably first-class performance that abides. The Cleveland Agreement does provide that "average" workers are to be studied in setting production standards. But we seem to make progress at times without being too meticulous in our interpretation of words and phrases. The life that can be read into such a document as the Cleveland Agreement is what counts.

Every system of wage remuneration—even the best of them—the author supposes, is open to abuse. Certainly under most of those in current use we can find outstanding instances of where the services of individuals which appear to be of equal value to the enterprise are differently compensated or where the same compensation is paid for services obviously differing in value. In his discussion of a paper read before the Taylor Society by Mr. William Green in December, 1925, Col. Sanford E. Thompson gave instances of this kind occurring under a piece-rate system in a shoe factory. Out of the practice of any industrial engineer similar examples may be cited almost without limit. The morale of any group—certainly its sense of solidarity—is seriously affected by such conditions. The group as well as the individual thrives on justice. If time study can be made one dependable means of securing a larger measure of justice in the relations between employer and employee, it will at the same time have removed a frequent source of friction—generally unrecognized—between the workers themselves.

Need for Better Formulation of Time-Study Standards for Benefit of Lay Public

It is hardly likely that time study will get strong group support except as it is of the very best variety made possible by the state of the art. Note this quotation from the Union comment on the Cleveland study:

> Mr. Goodell attempts to evaluate the data by four standards, namely, "Excellent," "Good," "Passable," and "Unsatisfactory." In our estimation, from the viewpoint of time-study data such as must be used if the standards system is to prove fair and satisfactory, there can be only two standards, namely, "Accurate" or "Reliable" (corresponding to Mr. Goodell's "Excellent") and "Inaccurate" or "Unreliable" (corresponding to his three other standards). There can be no grades in between. It is just because so much of the Cleveland time-study data is in the "Inaccurate" or "Unreliable" class, to say nothing of the careless way it is frequently used (all as revealed by Mr. Goodell's report), that trouble and suspicion result.

This seems to afford a further argument—if such be needed—for the organization of time-study practitioners to the end that some generally applicable standards of practice be set up and made available to the lay public.

We no longer associate with time study that simple, undifferentiated variety of exactitude which it was formerly assumed to possess. This observation is not intended to reflect in any way on the present or ultimate possibilities of the art. It is rather as a word of caution to those who in the first flush of enthusiasm as to an exceedingly interesting kind of work may easily fail to reckon with all the variants. It is a field wherein it pays to recall that the moon is not just where it was supposed to be, and that even the mathematicians are at a loss to know why—and this after our best minds have been giving plenty of attention to astronomy for several thousand years. Exactitude is a relative term—especially in the early days of any art or science.

In this connection the author recalls a statement of the late Col. Keppele Hall—a member of these Societies—to the effect that in a certain plant in Baltimore when there "was trouble on"—some dispute between the men and the management—the output on an operation with which he was specially familiar would drop 50 per cent and he—Colonel Hall—be unable to locate a single untoward act on any one's part. Without citing either complexes or inhibitions as possibly responsible, every industrial engineer has had the experience of setting what were considered fairly stiff standards only to see the workers perform them in a part of the time allowed.

In suggesting that in the past we have viewed time study as too exclusively a mechanism, the author hopes he will not be misunderstood. Our greatest chance of making time study an increasingly useful practice lies in the direction of making our observations more and more in detail and in seeking a greater and greater degree of precision in our conclusions. And yet when all this is said and done we shall fail, and deserve to fail, if we view time study exclusively in its mechanistic aspects. In the end it is invariably discovered that devices in themselves do not produce. Effectiveness comes here as everywhere, as a result of pride and joy in the job. Anything which does not further such ends we must view with suspicion.

If we had practiced time study over a much longer period than has actually been the case, and if, further, we had spent as much effort in winning the group to its support as in convincing the individual, we should still have to admit that the sum total of the ways in which a worker is capable of being influenced by the most accurate time studies represents only a part—perhaps a relatively small part—of the whole sum of influences which affect his conduct and output. Industry to be effective must be conceived as a process itself within a process. And to ascribe to time study anything more than the importance of one agency operating within this stream is to see it out of perspective.

Of course, on any such quest as this we have the future very largely in mind. We might easily decide that as affecting the present, time study does not play a role important enough to warrant discussion, and yet see in it a mechanism of very great importance in our study of industrial tendencies. Our problem in these Societies of course is to use present methods and present outlook only as a platform from which to build a better and more genuinely effective future. Can time study standardized as now practiced—or as its present practice may be revamped—be made to minister to the collective as well as the individual well-being of those engaged in industry and thus be made to increase industry's contribution to Society? That is the question we are discussing.

Perhaps from the standpoint of the group as contrasted with the individual the starting point in any discussion of time study should be a consideration of the conditions under which it can be held to be desirable. What are the earmarks of an appropriate setting for the profitable utilization of time-study methods? One rather bald answer to this inquiry is that for a stop watch by itself—unrelated by an approved procedure and technique to the rest of the industrial structure and process—there is no place. But to put the question in the way of any enduring settlement is the work of the group, not of the individual.

The most conspicuous characteristic of our current American industrial situation lies in the fact that individually and collectively we are increasingly awake and aware. The fact that we are somehow industrially on the move with a destination not yet fully determined, more and more fires the industrial imagination. In fact, we have become so accustomed to the idea of change that anything which appears static challenges suspicion. As an undercurrent to all this runs the constantly deepening conviction that we have only begun to tap the possibilities of production. The idea gradually grips us that only as the various factors in industry coöperate can we markedly raise the general standard of living. And with the understanding that there shall be a reasonable division of the returns from each recurring raising of the level of productivity, the workers are generally evidencing a keener interest in uncovering every possible source of waste—in material, in process, in management, and in individual and group conduct.

In the light of all this is it becoming more and more possible to bring any suggested policy—such as the establishment of production standards through time study—under a dispassionate and even friendly review by the group. But it is also becoming more and more evident that time study can play but a comparatively insignificant role where group morale has not been established. And further, if time-study analysis is to be given a recognized place in the evolving industrial régime, it will be largely because in its practice every proper consideration is given to its bearing on group morale.

Discussion

John A. Fitch[1] contributed a written discussion in which he said that on reading Mr. Cooke's paper, his first reaction had been that here was an extension of the ideals of scientific manage-

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ment in the true spirit of Frederick W. Taylor. To verify that impression he had turned to Taylor's Principles of Scientific Management, and found the following:

The writer is one of those who believes that more and more will the third party (the whole people), as it becomes acquainted with the true facts, insist that justice shall be done to all three parties. It will demand the largest efficiency from both employers and employees. It will no longer tolerate the type of employer who has his eye on dividends alone, who refuses to do his full share of the work and who merely cracks his whip over the heads of his workmen and attempts to drive them into harder work for low pay. No more will it tolerate tyranny on the part of labor which demands one increase after another in pay and shorter hours, while at the same time it becomes less instead of more efficient.

And the means which the writer firmly believes will be adopted to bring about, first, efficiency both in employer and employee and then an equitable division of the profits of their joint efforts, will be scientific management, which has for its sole aim the attainment of justice for all three parties through impartial scientific investigation of all the elements of the problem.

Mr. Cooke's paper was a record of the results of the sort of patient inquiry that Mr. Taylor had suggested,[2] and which was of the essence of scientific management. It was as scientific a method as the series of studies leading up to the discovery of high-speed steel, and if Mr. Cooke was less bold in his conclusions than Mr. Taylor had been at the end of that famous series of experiments it was because his materials were human and not inanimate and therefore subject to greater and less predictable variations.

There were two major obstacles to the achieving of that degree of coöperation between management and men that Mr. Taylor had advocated and that Mr. Cooke by implication favored in his paper. One of them had been recognized so clearly by Mr. Taylor that in a sense it had formed the basis of his philosophy and led him to the elaboration of what came later to be known as the principles of scientific management. This was the fact that the ideas and practices of management might be such as not only to make coöperation unlikely but such as to make it positively dangerous for the workers to attempt to adjust themselves consciously and enthusiastically to the purposes of management.

Nowhere, Mr. Fitch thought, would one find a more eloquent condemnation of restriction of output on the part of the workers nor a clearer indictment of management on account of its responsibility for it than in the Principles of Scientific Management.

The other obstacle was one that was clearly suggested by Mr. Cooke's paper. It was not enough that the purposes of management should be such as to entitle it to the confidence and support of the workers—they must have convincing evidence that such was the case. Wherever the rank and file of the workers were in ignorance of the problems and purposes of management, coöperation was unlikely and suspicion was to be expected.

Hence the great contribution to sound principles of management suggested in this paper. The clear inference to be drawn from the author's study of coöperation in a limited field was, Mr. Fitch thought, that the following steps should be taken: First the organization of the employees in any establishment on such a basis as to make possible the selection of such representatives to deal with management as would command the confidence and respect of the rank and file of the workers. Mr. Cooke suggested that this might be accomplished either through a trade union or through what had come to be known as a company union. Personally, Mr. Fitch was inclined to doubt that it could in the long run be accomplished through the agency of the company union. But he was in thorough agreement with Mr. Cooke's basic contention at this point. The prerequisite was that the representation must be such as to continue to command the confidence and respect of the rank and file of employees. If only that could be accomplished, the name given to it was immaterial.

The second essential was the frank interchange of opinion. It was essential that management and workers shall understand each other. If there was honesty of purpose on the part of management and a willingness to deal justly with employees, that fact would become increasingly apparent as the barriers of secrecy and mystery were removed.

The third essential suggested was that questions about which management and workers were unable to agree should be referred to the judgment of disinterested and intelligent outsiders.

[2] Taylor, Frederick W., The Principles of Scientific Management, pp. 138–139.

Mr. Cooke's paper was of outstanding importance because it showed that where these three principles had been in effect it had been possible to bring about coöperation with respect to one of the most difficult and knotty problems with which management and workers were concerned. In a field where suspicion generally ran riot, and where coöperation was usually withheld to the last ditch, it had been possible to develop understanding and coöperation to a very high degree. If such a feat could be accomplished in the field of production standards, were we not provided with a sound basis for the belief that much more extensive coöperation was possible?

Charles W. Lytle[3] and David B. Porter[4] contributed a joint discussion in which they wrote that Mr. Cooke had pointed out the only rational procedure for the introduction of time-study methods, namely, the treatment of employees as a group rather than individuals. The pioneers in this field had dealt with individuals because employers had always treated their workers that way. Today the specialization of machines and the extreme division of labor had broken down individual relationships and had welded the employees into groups which were keenly conscious of their common interests. This movement had manifested itself in many forms of employee representation and participation in management. It was wholly in accordance with modern developments, therefore, that such a vital factor as the establishments of production standards should be considered from the standpoint of its effect upon group reactions.

The Cleveland agreement showed an enlightened view on the part of the garment workers toward a mutual interest in lower unit costs, made possible by the measurement of output. On the other hand, the employers in that industry had shown great enlightenment in conceding to employees an equal interest in the determination of standards, and the recognition of the importance of steady work. Furthermore, the spirit of democracy in industry which was established by this agreement was thoroughly vindicated by the broad-minded and constructive criticism rendered by the garment workers through their union's letter of comment on Mr. Goodell's investigation of the standards situation. This kind of helpful participation is far removed from some early and now obsolete practices of carrying on time-study operations in secret.

Mr. Cooke had indicated that no group was going to be enthusiastic about a system of production standards which would bar a considerable portion of their number from further work. This group point of view compelled management's recognition. In order to satisfy this point of view some engineers had studied the normal worker, with the result that their standards, while seeming high at first, had been eventually too low. While interest should center around the normal worker, it was a mistake to neglect the "sport" or exceptional worker. His methods and motions might be studied and built into the structure of "the one best way." Thus his skill might be captured and acquired by the normal workers who ultimately raise their performance to a point approaching that of the exceptional worker. The increase was accomplished by the elimination of waste motion, not by additional energy. This had been illustrated in a recent paper before the American Management Association by Mr. Blakely of the General Electric Company, wherein he had cited the case of a worker whose performance had been below average. This person had been instructed in the methods of the one best way as developed by micro-motion studies, and ultimately had attained the performance of the fastest workers. The discussers wished to stress the educational aspect of capturing the "one best way" and teaching it to others. Advancement through greater earning power of course must follow. Mr. Cooke, they concluded, had stated at the beginning of his paper that improvement and standardization of processes was a necessary preliminary to final time study. They wished to emphasize the importance of that preliminary for two reasons: (1) because they believed that positive results had been defeated because of neglect of it, and (2) because in that phase lay many of the agencies for promoting the group morale so splendidly portrayed by the author.

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A Steam-Pressure Transformer

An Analysis of Koenemann's Recently Devised Process for Raising the Pressure of Exhaust Steam, and a Comparison Showing Its Apparent Advantages over the Mechanical Method Employing a Centrifugal Compressor

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IN ALL steam power plants the greater part of the heat coming from the boilers escapes ultimately in the exhaust steam. Usually this exhaust takes with it about 80 per cent of the heat supply, and as approximately 85 per cent of this heat is latent heat, about two-thirds of the heat supply to the engine is carried away as latent heat at the temperature of the exhaust. If the power plant is part of an industrial establishment requiring heat for process work, heating, etc., the exhaust may be utilized, but otherwise it is largely wasted. Similarly in many industrial processes, low-pressure steam is generated and wasted. This is particularly true in evaporation work, as in the sugar industry.

Various attempts have been made to utilize this low-pressure steam by raising it to a higher temperature and pressure and then employing it in some industrial process. The earliest suggestion of this kind was probably that of Lord Kelvin, who, in 1852, proposed the use of a reversed heat engine (a compressor) as a warming machine for air. He showed that, for a small temperature rise, the heat required by an efficient heat engine to drive the compressor is less than the heat which would be necessary for direct heating of the air. It is only in recent times, however, that this idea has been practically utilized for low-pressure steam, and its principal applications have been for evaporation processes—both single-stage and multiple-effect. For the economical generation of power this process cannot show any saving.

Compression Systems

The simplest practical procedure for the compression of low-pressure steam is by the use of a high-pressure steam jet acting on the low-pressure steam in an injector type of apparatus. This method has been extensively used in the Prache and Bouillon[2] evaporator. The injector type of compressor is inefficient. Tests[3] on an improved form of nozzle (DeBaufre) show, for example, that 1 lb. of steam at 125 lb. gage pressure will compress 1 lb. of steam from atmospheric pressure to 7.5 lb. gage pressure. This is only about 25 per cent of what is theoretically possible.

A distinctly better procedure, but one which is more complicated and entails considerably higher first cost, is the use of a centrifugal compressor driven either by a steam turbine or an electric motor. Reciprocating compressors, in spite of their higher efficiency, are usually undesirable on account of their large volume and of the presence of lubricating oil. Centrifugal compressors have been used for many evaporators, and especially in those by Soderlund and Boberg.[4] Tests by Ombeck[5] on an evaporator, with vapor recompression by a centrifugal compressor operated by an electric motor, show the following typical results:

- *a* With pressure range from 0.25 lb. gage to 4.9 lb. gage, a compression of 76 lb. of vapor per kw-hr.
- *b* With pressure range from 0.41 lb. gage to 2.3 lb. gage, a compression of 165 lb. of vapor per kw-hr.

Assuming a steam consumption of 20 lb. per kw-hr., the steam compressions are (*a*) 3.8 and (*b*) 8.25 lb. per lb. of high-pressure steam. With a high pressure of 200 lb. abs., the theoretical yield (Rankine cycle) of steam at the compression pressure (= weight of low-pressure steam compressed plus expanded high-pressure steam) is (*a*) 10.7 lb., (*b*) 26 lb. per lb. of high-pressure steam. The efficiencies are (*a*) 35 per cent, (*b*) 32 per cent, as compared with 25 per cent found with the injector type.

The Koenemann Chemical System

An entirely different procedure has recently been devised by Dr. Ernst Koenemann of Berlin, who proposes to use chemical in place of mechanical methods for compressing the exhaust steam. The object of this note is to present an analysis of the Koenemann process. In this process, use is made of the fact that the boiling point of a solution (e.g., of KOH or NaOH in water) is higher than that of water at the same pressure. If exhaust steam, at atmospheric pressure, is blown into a solution of KOH of a concentration having a boiling point of 300 deg. fahr., and if the solution is already at the temperature of 300 deg. fahr., the exhaust steam (at 212 deg. fahr.) will be condensed by the higher-temperature solution and will dilute that solution. The latent heat of the exhaust steam will be liberated and also the heat of dilution, and except for the heat used in raising the condensed steam from 212 deg. to 300 deg. fahr., this liberated heat will be available at 300 deg. fahr. and can be utilized for generating steam at that temperature and at the corresponding saturation pressure (67 lb. per sq. in. abs.). In order to make the process continuous, the diluted solution must next be reconcentrated, which may be accomplished in another vessel, by high-pressure-steam coils or other source of heat. If the pressure in this other vessel is maintained at 67 lb. abs., the steam evaporated can be mixed with the steam generated in the first vessel without loss.

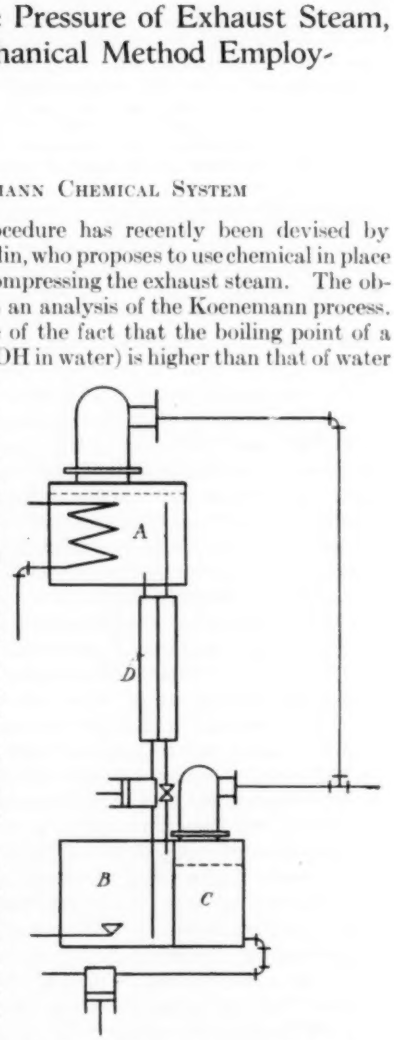

Fig. 1 Diagrammatic Sketch of Koenemann's Apparatus

Koenemann's Apparatus and the Theory of Its Operation

A diagrammatic sketch of the apparatus required is shown in Fig. 1. Low-pressure steam is blown into the vessel *B* which is full of a solution of KOH at the pressure of the low-pressure steam. The heat liberated passes by conduction to water in the vessel *C* and evaporates it at the saturation pressure corresponding to the temperature in *B*. This will be called the *intermediate pressure*. To maintain the concentration in *B* constant, the potassium hydroxide solution in *B* is pumped through the heat exchanger *D* into the vessel *A* (which is at the intermediate pressure), is reconcentrated by the steam coil shown, and returns to *B*, passing through the heat exchanger *D* and a throttle valve. The steam generated in *A* joins the steam coming from *C*. A feed pump supplies fresh water to *C*.

Consider the energy transfers when an infinitesimal quantity of water dx is added to the vessel *B* in which a constant temperature is maintained (Fig. 2). The strength of the solution c will be defined as $\dfrac{KOH}{KOH + H_2O}$. Suppose $(1 - dx)$ lb. of strong solution of concentration c returned from *A* while dx lb. of water is added, forming 1 lb. of weak solution of concentration $c - dc$. As the

[1] Professor of Mechanical Engineering, Harvard University. Mem. A.S.M.E.
[2] *The Engineer*, vol. 102, pp. 637–640 (1916).
[3] *Chemical and Metallurgical Engineering*, vol. 28, pp. 26–31 (1923).
[4] *Journal of the Society of Chemical Industry*, vol. 36, pp. 70–72 (1917).
[5] *Chemical and Metallurgical Engineering*, vol. 28, pp. 26–31 (1923).

concentration $c - dc$ is obtained by adding dx of water to the solution of concentration c, we have

$$c - dc = \frac{c(1-dx)}{c(1-dx)+(1-c)(1-dx)+dx} = c - cdx$$

or

$$dc = cdx \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

Let h = total heat per lb. of the water added to B
h_c = total heat per lb. of the strong solution
$h_c - dh_c$ = total heat per lb. of the weak solution
y = heat of dilution, defined as the heat liberated when 1 lb. of water is added to an infinite quantity of solution of strength c, and
dQ = the heat set free by the reaction in vessel B under the conditions of Fig. 2.

Then, considering all the energy entering and leaving the vessel B,

$$hdx + (1-dx)h_c = h_c - dh_c + y_c dx$$

$$(h - h_c - y_c)dx + dh_c = 0$$

and

$$\frac{dh_c}{dx} = h_c + y_c - h$$

From Equation [1]

$$c\frac{dh_c}{dc} = h_c + y_c - h$$

or

$$\frac{dh_c}{dc} - \frac{h_c}{c} - \frac{y_c}{c} + \frac{h}{c} = 0 \quad \ldots\ldots [2]$$

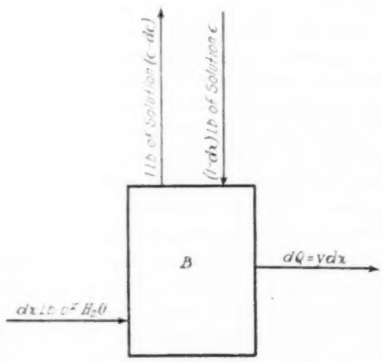

FIG. 2 ENERGY TRANSFERS WHEN AN INFINITESIMAL QUANTITY OF WATER dx IS ADDED TO VESSEL B IN WHICH A CONSTANT TEMPERATURE IS MAINTAINED

From the data available on the heat of dilution of KOH we may write

$$y_c = 1.8 \times (10c)^{3.17} \text{ B.t.u. per lb.} \ldots\ldots [3]$$

Consequently

$$\frac{dh_c}{dc} - \frac{h_c}{c} - (1.8 \times 10^{3.17}c^{2.17}) + \frac{h}{c} = 0$$

and

$$h_c = 1.8\frac{10^{3.17}}{2.17}c^{3.17} + h + Kc, \text{ B.t.u. per lb.} \ldots\ldots [4]$$

where K is a constant of integration.

Now consider the process with steam added to B and with finite differences of concentration (Fig. 3). In a given period of time the weight w_0 of steam enters B, a weight w_2 of strong solution of concentration c_2 is returned from A to B, and a weight $w_1 = w_2 + w_0$ of weak solution of concentration c_1 goes from B to A. By the action of the heat interchanger D, the temperature of the strong solution entering B is the same as that of the weak solution leaving B. If h_0, h_1, and h_2 are the total heats per pound of the steam, weak solution, and strong solution, respectively, then

$$w_0h_0 - w_1h_1 + w_2h_2 - Q = 0 \ldots\ldots\ldots\ldots [5]$$

where Q is the heat going to C.

Also

$$c_1w_1 = c_2w_2 = c_2(w_1 - w_0)$$

or

$$\frac{w_1}{w_2} = \frac{c_2}{c_2 - c_1} \ldots\ldots\ldots\ldots [6]$$

From [5] and [6],

$$\frac{Q}{w_0} = h_0 - \frac{c_2}{c_2-c_1}h_1 + \frac{c_1}{c_2-c_1}h_2$$

or

$$\frac{Q}{w_0}(c_2 - c_1) = (c_2-c_1)h_0 - c_2h_1 + c_1h_2$$

$$= c_2(h_0 - h_1) - c_1(h_0 - h_2)$$

Now $h_0 = h + L_0$, where h is the heat of the liquid and L_0 is the latent heat of the low-pressure steam. And from [4],

$$h_1 = \phi(c_1) + h + Kc_1$$

and

$$h_2 = \phi(c_2) + h + Kc_2$$

where

$$\phi(c) = 1.8\frac{10^{3.17}}{2.17}c^{3.17}$$

$$\therefore \frac{Q}{w_0}(c_2 - c_1) = c_2L_0 - c_2\phi(c_1) - Kc_1c_2 - c_1L_0 + c_1\phi(c_2) + Kc_1c_2$$

$$= (c_2 - c_1)L_0 + [c_1\phi(c_2) - c_2\phi(c_1)]$$

and

$$\frac{Q}{w_0} = L_0 + \frac{c_1}{c_2-c_1}\phi(c_2) - \frac{c_2}{c_2-c_1}\phi(c_1)$$

$$= L_0 + \frac{c_1c_2}{c_2-c_1}\,\frac{1.8\times 10^{3.17}}{2.17}(c_2^{2.17} - c_1^{2.17}) \text{ B.t.u. per lb.} \ldots [7]$$

Equation [7] gives the quantity of heat available for transmission to the water in vessel C per pound of low-pressure steam blown into B. If the total heat of the saturated steam generated in C is H, and if C is supplied with feedwater at the temperature of the low-pressure steam, the weight of steam generated in C per lb. of low-pressure steam is $w = \dfrac{Q}{w_0(H-h)}$ lb.

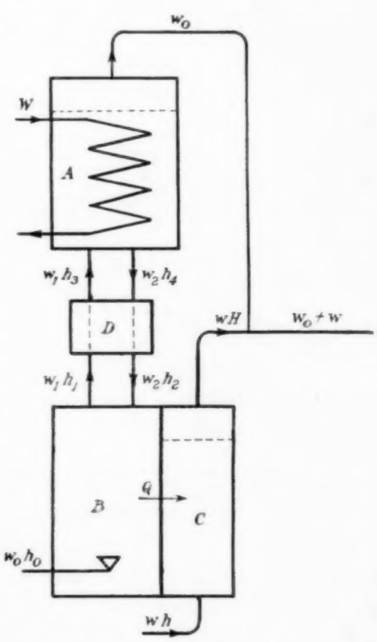

FIG. 3 PROCESS WITH STEAM ADDED TO B AND WITH FINITE DIFFERENCES OF CONCENTRATION

The temperature in B (and consequently the pressure in C) is determined by the concentration of the solution. At atmospheric pressure the relation between boiling temperature and concentration is given by Fig. 4. At other pressures an approximate relation may be obtained by assuming that the ratio of the boiling temperatures (absolute) at any two pressures is the same as for steam at the same two pressures.

The heat exchanges in the vessel A may be obtained by the same general procedure as in vessel B. With a perfect heat exchanger D,

$$w_2h_4 - w_2h_2 = w_1h_3 - w_1h_1 \quad \ldots\ldots [8]$$

where h_4 is the total heat per pound of the strong solution leaving A, and h_3 is the total heat per pound of the weak solution entering A. The pressure of the high-pressure steam must be that corresponding to the saturation temperature of the strong solution at the pressure of the steam generated in C. The steam generated in A will have the same pressure as in C, but will be superheated up to the temperature of the high-pressure steam. The weight evaporated in A must be the same as the weight condensed in B.

Let W = weight of high-pressure steam condensed in A for w_0 lb. condensed in B
L = its latent heat, and
H_T = total heat per pound of the superheated steam generated in A measured above the heat of the liquid at saturation temperature.

Then

$$WL + w_1h_3 - w_2h_4 - w_0H_T = 0$$

From [8]

$$WL + w_1 h_1 - w_2 h_2 - w_0 H_T = 0$$

and

$$\frac{W}{w_0} L = H_T + \frac{w_2}{w_0} h_2 - \frac{w_1}{w_0} h_1$$

$$= H_T + \frac{c_1 c_2}{c_2 - c_1} \frac{1.8 \times 10^{3.17}}{2.17} (c_2^{2.17} - c_1^{2.17}) \text{ B.t.u. per lb.} \quad [9]$$

This equation gives the weight of high-pressure steam required per pound of low-pressure steam condensed. The above equations assume ideal conditions, with no temperature differences between the bodies exchanging heat by conduction, with a perfect heat interchanger, and neglecting the small amount of pump work.

The results of calculations of the performance of the Koenemann system are given in Table 1. A value of $c_2 - c_1 = 0.05$ is assumed in each case except the last but one, in which $c_2 - c_1 = 0.01$. The temperatures in the vessels B, C, and A are given in columns 3 and 4; the pressure in A and C is given in column 6; and the pressure in the high-pressure steam coils in column 5; the pressure in B is assumed to be 14.7 lb. abs. The weight of steam generated in C per pound of exhaust steam condensed in B is given in column 7; the corresponding weight of high-pressure steam condensed in the coils is given in column 8. The total weight of steam generated is the sum of 1 lb. of superheated steam generated in A and the quantity given in column 7. If this is divided by the quantity in column 8, it gives the values of column 9. The final column gives the ratio of the heat required to generate the intermediate-pressure steam actually formed, to the heat supplied to generate the high-pressure steam, both measured from feedwater at 212 deg. fahr. For example, in the last line of Table 1, if steam is required at 28 lb. abs. it would require 1.84 times as much heat to generate it directly in a boiler as would be required to generate it by use of the KOH system employing high-pressure steam at 58 lb. pressure.

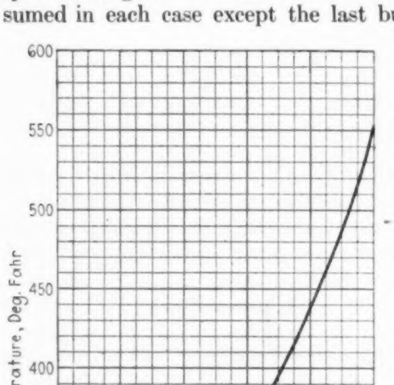

FIG. 4 RELATION BETWEEN BOILING-TEMPERATURE AND CONCENTRATION OF KOH SOLUTION AT ATMOSPHERIC PRESSURE

TABLE 1 CALCULATED PERFORMANCE OF KOENEMANN SYSTEM, WITHOUT LOSSES

Concentrations c_1 (1)	c_1 (2)	Temperatures, deg. fahr. In B and C (3)	In A (4)	Pressures, lb. per sq. in. abs. of h.-p. steam (5)	of steam generated in A & C (6)	Steam generated in C per lb. of l.-p. steam, lb. (7)	H.-p. steam condensed per lb. of l.-p. steam, lb. (8)	Total steam generated per lb. of h.-p. steam condensed, lb. (9)	Ratio of total heat of steam generated to total heat of h.-p.steam, measured above 212 deg. fahr. (10)
0.6	0.55	322	487	600	92	1.43	1.94	1.25	1.26
0.55	0.5	293	417	300	60	1.32	1.63	1.43	1.42
0.5	0.45	270	360	153	42	1.23	1.42	1.57	1.57
0.45	0.4	253	315	83	31	1.17	1.28	1.70	1.69
0.41	0.4	253	308	75	31	1.14	1.24	1.73	1.72
0.4	0.35	246	290	58	28	1.10	1.14	1.84	1.84

COMPARISON OF MECHANICAL AND CHEMICAL SYSTEMS

A comparison of the performance of the Koenemann system with a compression system is given in Table 2. The same steam pressures are taken as for the Koenemann system. The high-pressure steam is assumed to be used, without loss, in an engine using the Rankine cycle and exhausting at the intermediate pressure and consequently at the quality given in column 4. The low-pressure steam is assumed to be compressed to the intermediate pressure in a reversed Rankine cycle, and consequently is discharged with the superheat given in column 6. The total weight of intermediate-pressure steam is the sum of the dry steam in the exhaust from the engine and the superheated steam discharged from the compressor. The ratio of the heat that would be required to generate the total weight of intermediate steam, to the heat supplied to generate the high-pressure steam (both measured from feedwater at 212 deg. fahr.), is given in column 7. The comparison of this quantity with the similar quantity for the Koenemann system is given in column 8, Table 2. It shows that the Koenemann system, without losses, is less efficient than the compression system without losses, but the differences are not great when the intermediate pressures are moderate.

TABLE 2 COMPARISON OF PERFORMANCE OF KOENEMANN SYSTEM WITH COMPRESSION SYSTEM

		Compression System					
Concentrations c_1 (1)	c_1 (2)	Expansion work per lb. of h.-p. steam, B.t.u. (3)	Quality of h.-p. steam after expansion (4)	Compression work per lb. of l.-p. steam, B.t.u. (5)	Superheat of l.-p. steam after compression, deg. fahr. (6)	Ratio of total heat of compressed steam to total heat of h.-p. steam, measured above 212 deg. fahr. (7)	Ratio of performance of Koenemann system to that of compression system = (Col. 10, Table 1) ÷ (Col. 7, Table 2) (8)
0.6	0.55	147	0.865	161	261	1.85	0.68
0.55	0.5	126	0.894	122	192	1.97	0.72
0.5	0.45	100	0.918	88	138	2.06	0.77
0.45	0.4	75	0.94	60	95	2.20	0.78
0.41	0.4	67	0.946	60	95	2.08	0.83
0.4	0.35	55	0.956	51	82	2.05	0.90

The theoretical performance of the compression system cannot be closely approximated to in practice; the efficiency of both turbine and centrifugal compressor will usually not exceed 60 to 70 per cent, or a combined efficiency of about 40 per cent. The tests quoted at the beginning of this paper show a combined efficiency of about 35 per cent.

In the Koenemann system the actual performance may be expected to approximate fairly closely to the theoretical performance. The necessity for temperature difference between B and C, and between the steam coils and A, will not make a noteworthy change in efficiency; the heat-interchanger loss also may be kept quite small. With a theoretical efficiency of 90 per cent of the compressor system, and with 80 per cent realization of the theoretical efficiency, the actual efficiency would be about 70 per cent of the theoretical efficiency of the compression system, or, nearly twice the actual efficiency of the compression system.

A comparison of the compression and the Koenemann systems appears to promise three advantages for the latter: (*a*) higher efficiency, (*b*) the absence of moving parts except for a small pump, and (*c*) the possibility of obtaining steam at pressures considerably higher than are practicable with a centrifugal compressor, and consequently better adaptability for use in multiple-effect evaporators.

Improvements in Die Castings

TEN years ago the die-casting process was thought suitable only for comparatively small castings; but gradually their size has increased until now it is possible to successfully produce aluminum die castings up to 10 lb. in weight, and zinc castings more than twice as heavy.

Recently we had an opportunity to inspect the records of tests made on an improved zinc-base die-casting metal, which had a tensile strength ranging from 48,000 to 50,000 lb. per sq. in., with an elongation varying from 3.5 to 6.5 per cent. In addition the test specimen proved to have remarkable resistance to twisting strains.

Chromium plating is being experimented with for producing a hard wearing surface on the soft die-casting metal, and if this process can be applied to die castings, there is no reason why they should not wear as well as case-hardened steel.—*Machinery*, May, 1927, p. 658.

Present Tendencies of Steam-Station Design

Improvements Since 1913—Higher Steam Pressures and Temperatures—Use of Economizers—Effect of Stage Bleeding for Feedwater Heating on Turbine Design—Use of Air Preheaters—Furnace Design—Improved Methods of Burning Coal—Increase in Size of Equipment—Future Possibilities

By VERN E. ALDEN,[1] BOSTON, MASS.

IN ORDER to obtain an understanding of the present tendencies of steam-station design, it is necessary to review the progress made in recent years.

The underlying motive which has spurred power-station engineers on has been the desire to build steam generating stations to obtain the following results:

1 A high degree of reliability in operation
2 Low fuel cost per kilowatt-hour of net station output
3 Low cost of operating labor
4 Low maintenance cost
5 Low investment and corresponding low fixed charges per kilowatt-hour of station output
6 That the station as designed and built shall be so well fitted for the work which it shall be called upon to perform, not only during the first year of its operating life, but also during the later years of its life, that it shall be unnecessary to undertake any major program of rebuilding in the future.

The extensive programs of steam-station construction made necessary by the growth of loads of the electric light and power companies have given the power-station engineers ample opportunities to develop improved designs and to test these designs quickly by actual operation.

Improvements Since 1913

The rapidity with which improvements have been developed has been rather startling. The extent of the improvement in fuel economy is indicated by the curve in Fig. 1, which represents the weighted average of the coal consumption in pounds per kilowatt-hour of net station send-out for a number of typical stations of 60,000 kw. capacity and higher plotted as ordinates, with the dates of initial operation for the different stations as the abscissas. The operating results for the different stations have been reduced to a common basis as regards quality of fuel, coal having been assumed as having 14,200 B.t.u. per lb. as fired. It is to be noted that in 14 years' time the coal consumption for typical stations has been reduced from 1.6 to 0.9 lb. per kw-hr.

This improvement in operating performance appears even more remarkable when it is remembered that there are in this country quite a number of steam generating stations of 50,000 kw. capacity and higher, in connection with which the coal consumption is from 2 to 2.4 lb. per kw-hr. of net station send-out.

While the stations taken as a basis for plotting the curve in Fig. 1 are referred to as typical stations, it should be borne in mind that the two stations having coal consumptions below 1 lb. per kw-hr. are not entirely typical, as a number of well-designed stations which have gone into operation during 1926 have coal consumptions somewhat in excess of 1 lb. per kw-hr. It should be clearly borne in mind that it is only in connection with stations which will be operated for many years at a high load factor and in connection with which the cost of fuel in cents per million B.t.u. is high that the refinements in design necessary to produce a kilowatt-hour for one pound of coal or less can be justified.

The number of men required for the operation of the typical station placed in operation in 1926 is less than one-half that required for the operation of a typical station of equal capacity placed in operation in 1913. The maintenance forces have been reduced by at least 40 per cent for the typical station. The increases in the wage scale have, however, more than offset these reductions in the amount of labor required for shift operation and maintenance, so that actually the costs of operating labor and maintenance in mils per kilowatt-hour of station send-out have increased.

In order to understand the nature of the changes which have made it possible to effect these sweeping reductions in coal consumption and in the amount of labor required for operation and maintenance of large generating stations, it is necessary to picture the typical steam generating station of 1913.

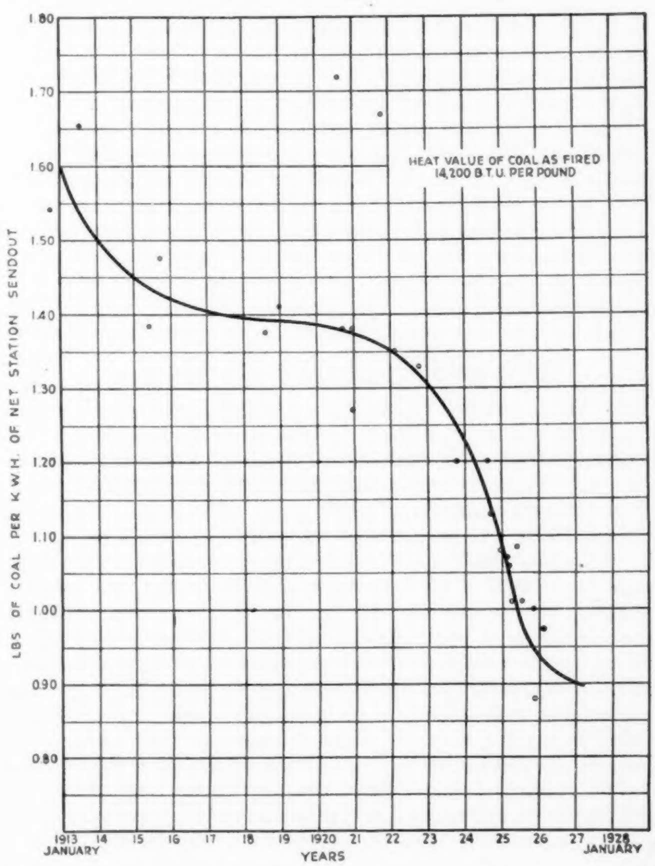

Fig. 1 Performance of Typical Stations Plotted Against Date of Initial Operation of Station

Typical Station of 1913

Such a station would have had a capacity of, say, 45,000 kw., this capacity being made up of three 15,000-kw. steam turbines. With each of these 15,000-kw. steam turbines there would have been installed a 27,000-sq. ft. two-pass surface condenser arranged for the circulation of approximately 38,000 gal. of condensing water per minute. The air would have been withdrawn from the condenser by means of one or more air pumps of the hurling-water type. In the boiler house there would have been installed 18 straight-tube, longitudinal-drum boilers, each having an area of 10,000 sq. ft. Installed in connection with each of these boilers there would have been a convection-type superheater at the top of the tube bank. Steam would have been delivered at the boiler stop valve at a pressure of 190 lb. per sq. in. gage and a total tem-

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Contributed by the Power Division and presented at the Kansas City, Mo., Meeting, April 4 to 6, 1927, of The American Society of Mechanical Engineers.

perature of 500 deg. fahr. Each boiler would have been fired by means of a 10-retort, standard-length, 17-tuyere stoker of the underfeed type with hand-operated dump grates. The combustion chamber would have had a total volume of approximately 1300 cu. ft., being 18 ft. wide by 9 ft. deep with a total average height of 8 ft. Forced-draft fans for the stokers and natural-draft chimneys would have been employed. For continuous operation for extended periods of time the output of the boilers would have been limited to 200 per cent of their normal rating. If operated at this maximum rating for any appreciable percentage of the operating hours, extended outages for the repair of the brick settings would have been required. The forced-draft fans and most of the other auxiliaries in the boiler house and turbine room would have been driven by means of steam turbines. In connection with the circulating pumps and the separate exciter, dual drive by means of motor and turbine would have been provided for reliability and in order roughly to control the amount of the exhaust steam required for feedwater heating. All of the steam turbines for driving the auxiliaries would have exhausted into a common header against approximately atmospheric back pressure. This exhaust steam would have been used in open feedwater heaters for heating the condensate pumped direct to these feedwater heaters by means of the condensate pumps. The air for cooling the generators would have been drawn from outside to the turbine room, passed through an air washer, and perhaps discharged through an air duct into the basement of the boiler house. Each boiler would have been operated as a separate unit and but a scanty complement of instruments would have been provided in the boiler room for the guidance of the firemen in maintaining proper conditions of combustion.

We may now trace the changes which have taken place.

Changes Since 1913—Steam Pressure

One obvious way of improving steam-station economy is to make it possible to convert a larger percentage of the total heat in each pound of steam into mechanical energy by increasing the steam pressure. In 1915 one or more stations went into operation at a steam pressure of 225 lb. In 1916 the River Station of the Buffalo General Electric Company went into operation at 275 lb. pressure. In 1917 the Joliet Station was placed in operation with a steam pressure of 300 lb. In 1920 the Springdale Station was placed in operation at a pressure of 350 lb. In 1924 the Trenton Channel Station in Detroit was placed in operation at a pressure of 415 lb. per sq. in. In 1925 the Philo, Twin Branch, and Crawford Avenue Stations were placed in operation at a pressure of 550 lb. per sq. in. and early in 1926 a 3000-kw. turbine and a single large boiler were placed in operation in the Edgar Station in Boston at a steam pressure of 1200 lb. per sq. in. At the present time in the Edgar Station there are being installed two new boilers and a 10,000-kw. turbine, which will operate as a reducing valve between the boilers and the throttle of a 65,000-kw. turbine, and these boilers and their high-pressure turbine are to operate at a steam pressure of 1400 lb. per sq. in.

Steam Temperature

It was further recognized that a larger percentage of the total heat in each pound of steam leaving the boiler could be converted into mechanical energy at the turbine coupling if the steam temperature could be increased. This comes about by virtue of improved cycle efficiency and is also due to the fact that the blade and disk friction losses in the turbine are reduced as the result of less moisture in the steam, this making it possible for the turbine to perform more nearly as a perfect turbine should.

In 1915 the Connors Creek Station in Detroit went into operation with a steam temperature of 600 deg. fahr. In 1916 the River Station of the Buffalo General Electric Company went into operation with a steam temperature of 690 deg. fahr. In the years that followed a number of stations were operated with steam temperatures ranging between 600 and 650 deg. fahr. and a working knowledge was obtained which served as the basis for the redesign of piping, valves, and fittings along lines which would eliminate the rather high maintenance encountered in connection with some of the earlier stations that employed steam at high temperatures.

It was recognized that some form of superheater other than the conventional convection type installed at the top of the tube bank, would have to be used if steam temperatures of 700 deg. fahr. or higher were to be easily obtained. It was further recognized that it would be necessary to maintain the steam temperature within a moderate range more or less independent of the load carried by the boiler, if steam temperatures of the order of 700 deg. fahr. were used. Two solutions were proposed:

First, the use of the interdeck superheater as embodied in the boilers of the Hell Gate Station in New York placed in operation in 1921. In these boilers a space is left between the upper and the lower portion of the main tube bank and the superheater is installed with 6 or 8 boiler tubes between it and the furnace, so that it is brought in contact with relatively hot gases.

Second, the use of the radiant-heat superheater as embodied in the boilers in the Lakeside Station in Milwaukee placed in operation in 1923. This radiant-heat superheater consists of elements installed in the furnace and forming a part of the walls. These elements absorb radiant heat directly from the fire, and it is possible by means of a relatively small amount of surface to absorb sufficient heat to raise the steam temperature to 700 deg. fahr. or higher.

With the experience gained in the period between 1916 and 1923 as a basis, and with the new types of superheaters available, it was possible to design the Philo and Twin Branch Stations for an operating steam temperature of 725 deg. fahr., and the latest units which have been installed in the Crawford Avenue Station in Chicago and the 94,000-kw. turbine for the Long Beach Station of the Southern California Edison Company are designed for a temperature of 750 deg. fahr.

Ferranti called attention many years ago to the possibility of improving cycle efficiency and reducing the friction losses in the low-pressure blading of large turbines by withdrawing the steam from the turbine after it had expanded part way, taking it back to the boiler room, reheating it to a value near its original temperature, and then returning the steam to the turbine to be completely expanded to the pressure at the exhaust nozzle. This feature was embodied in the North Tees Station in England placed in operation at a steam pressure of approximately 450 lb. about 1920. So little information in regard to this station was available and such rumors as came to us were so disquieting that there were no immediate steps taken in this country to follow up this new development. In 1925, however, the Philo, Twin Branch, and Crawford Avenue Stations were placed in operation, all the turbines operating with a single stage of reheating to a steam temperature of approximately 725 deg. fahr. There is at the present time, either in operation or in the course of construction, approximately 600,000 kw. of turbine capacity designed for operation with a steam pressure of 550 lb. per sq. in. gage and a steam temperature of approximately 725 deg. fahr. at the turbine throttle; all of this turbine capacity being designed for a single stage of reheating to a steam temperature of approximately 725 deg. fahr. Reheating of the steam is also employed in connection with the 1200-lb.-pressure installation in the Edgar Station.

Heating of Feedwater and the Use of Economizers

It was early recognized that feedwater should be heated to a temperature higher than the 210 deg. fahr. possible in connection with an open feedwater heater taking exhaust steam from the auxiliary turbines at atmospheric pressure. It was further recognized that the heat in the flue gases leaving the boiler might be used for heating the feedwater. The first step, therefore, was to use economizers extensively for reducing the temperature of the flue gases and increasing the temperature of the feedwater.

The use of higher steam pressures complicated the problem of economizer design. The use of economizers with cast-iron elements for an operating pressure of 300 lb. or higher seemed undesirable. One solution embodied in several stations placed in operation in 1919 was to use a cast-iron economizer designed for an operating pressure of approximately 200 lb., the feedwater being heated to a temperature of approximately 325 deg. fahr. in this economizer and then delivered to a second boiler-feed pump, which, acting as a booster, increased the pressure to the value necessary to deliver the water to the boilers. This arrangement so complicated the design of the piping and equipment that it was abandoned in

favor of the steel-tube economizer, which is suitable for operation at pressures of even as high as 1500 lb. per sq. in.

A few months' operating experience with steel-tube economizers developed pitting of drums, headers, and tubes. This pitting was comparatively rapid and would, had it continued, have resulted in prohibitive costs for replacements. It was found, however, that this pitting was in large measure due to the presence of dissolved oxygen in the feedwater delivered to the economizers, so that in practically all stations having steel-tube economizers provision was made for its removal. This solution effectively inhibited the pitting.

The use of economizers brought with it the extensive use of induced-draft fans, and with the removal of the natural-draft stack as a limiting factor, operating men began to investigate the possibilities of boiler operation at higher ratings than had previously been the fashion.

It had been recognized for many years that the heating of the feedwater in an open feedwater heater by means of steam exhausted from the auxiliary turbines was by no means an ideal solution. Even with the best type of small turbines available it was not possible to develop a horsepower-hour with less than 17 lb. of the steam which was used for feedwater heating. Power-station engineers who stopped occasionally to day-dream, recognized that it might be possible to bleed steam out of the main turbines at a number of different points for feedwater heating, and by this method to generate a very much larger amount of power with the quantity of steam which was required to heat the feedwater to a given final temperature.

The possibilities of increasing the amount of power generated by the use of steam bled from the turbine may be visualized when it is remembered that the water rates in connection with steam bled from the turbine at 5.5, 20, 76, and 195 lb. absolute pressure for feedwater heating are approximately 12.9, 17.5, 30.2, and 83 lb. per kw-hr., respectively. For heating the feedwater up to a temperature of 160 deg. fahr., steam at a pressure of 5.5 lb. per sq. in. absolute serves just as well as steam at a pressure of 195 lb. absolute. The amount of power generated by the steam used for feedwater heating will, however, be increased six times by the use of steam bled out of the turbine at the lower pressure.

This solution seemed somewhat too complicated, but almost over night power-station engineers came to realize that it had many advantages. The complications in design incident to the use of three or more heaters in connection with each turbine for feedwater heating were found to be more apparent than real. Operating experience proved that the heating of feedwater by stage bleeding actually simplifies station operation. Practically all of the power stations which have been placed in operation since 1923 have been designed for the regenerative cycle of operation with the feedwater heated by means of steam bled from the turbine at from two to five points.

Effect of Stage Bleeding for Feedwater Heating on Design of Turbines

A major advantage of the cycle in which steam is bled from the turbine at a number of points for feedwater heating is that it permits of increasing the output from a given size of turbine. The factor which fixes the overall proportions of a turbine designed for condensing operation with a vacuum of from 28 to 29 in. of mercury is the volume of steam which passes through the last row of moving blades. For a turbine from which steam is bled at four points for heating the feedwater to a final temperature of approximately 375 deg. fahr., the turbine receiving steam at a pressure of approximately 400 lb. per sq. in., only 75 per cent of the total weight of steam which enters the turbine throttle will pass all the way through to the condenser. The other 25 per cent will be bled out at absolute pressures of approximately 5.5 lb., 20 lb., 76 lb., and 195 lb., respectively, for feedwater heating. The overall dimensions of the turbine are fixed by the volume of the steam which, passing through the last row of moving blades, is exhausted to the condenser. The remaining weight of steam which passes only part way through the turbine does not to any appreciable extent increase the overall dimensions or the cost of the turbine.

This fact may be more clearly visualized when it is borne in mind that steam entering the turbine throttle has a volume of approximately 1.6 cu. ft. per lb., whereas the steam passing through the last row of moving blades has a volume of approximately 600 cu. ft. per lb. However, the output of the turbine is increased by the power generated by the steam which is expanded only part way through the turbine, and this increase amounts to approximately 13 per cent.

It is being recognized that an outstanding advantage of the regenerative cycle as worked out for four- or five-stage bleeding for feedwater heating is the overload capacity of the turbine unit which is made available. In approximately all large systems the peak demand occurs during the month of December, and corresponding to the time of this peak demand the condenser circulating water is at a temperature of approximately 40 deg. fahr. With these conditions of operation it will be possible, by accepting a moderate increase of approximately 6 per cent in the coal consumption, to carry a total load at least 10 per cent in excess of the maximum load which it is possible to carry on the turbine unit under normal operating conditions. This increase in the capacity of the turbine unit is effected by closing the valves in the lines leading to the two top-stage feedwater heaters. Steam which under normal conditions of operation is bled out of the turbine into these heaters then passes through the low-pressure stages of the turbine, increasing the output of the machine. The slight increase in coal consumption comes as a result of the reduction in the feedwater temperature from approximately 375 to 225 deg. fahr. incident to the removal of the two top-stage heaters from service. Normal conditions of operation may be restored and the coal consumption per kilowatt-hour reduced as soon as the peak demand has passed, by opening the two valves in the bleeder lines. The low temperature of the water going to the generator air coolers corresponding to December operation, makes it possible to carry the overload on the generator with total temperatures which are actually lower than those corresponding to summer operation at normal rated load.

Use of Air Preheaters

The introduction of the regenerative cycle for feedwater heating brought with it a new problem. If the feedwater is heated by means of steam bled from the turbine to a temperature of from 300 to 375 deg. fahr., it is not possible by passing this feedwater through an economizer to pick up any appreciable amount of additional heat nor to effect any appreciable reduction in flue-gas temperatures.

If the coal consumption per kilowatt-hour of station send-out is to be further reduced, however, the amount of heat lost in the flue gases going up the stack must be kept to a reasonably low figure, and some form of heat trap must be installed between the boiler and the stack.

Air preheaters had been used to some extent abroad in marine work, and to a very limited extent in process work in this country. Air preheaters were installed during 1923 in the Colfax Station in Pittsburgh with underfeed stokers and in the Calumet Station in Chicago and the Northeast Station in Kansas City where low-grade coal was burned on chain-grate stokers. In all three cases operation showed the entirely unforeseen result of improved fuel-bed conditions, and with the chain-grate stokers burning low-grade Illinois coal it was actually possible to operate the boilers at higher ratings with the preheated air than with air supplied to the fuel bed at normal room temperature.

The first air preheaters were designed for a very moderate increase in air temperature and a moderate reduction in flue-gas temperature. In the stations placed in operation during the last year, however, the possibilities of the air preheater have been more fully realized. The gas velocities through the air preheater have been increased so as to increase the heat-transfer rate, and the temperature of air for combustion has been raised from room temperature to approximately 500 deg. fahr. with a corresponding drop in temperature of flue gases from approximately 725 deg. fahr. leaving the boiler to 350 deg. fahr. leaving the air preheater.

Particularly with stoker-fired installations there has been a trend toward the use of boilers from 18 to 22 tubes high in order to limit the temperature of flue gases at the boiler uptake. This in turn limits the temperature of preheated air delivered to the stoker to a more moderate value.

Furnace Design

As long as stokers of comparatively limited projected area were used and the natural-draft chimney set the limitation to boiler output, there was no necessity of going to furnaces of other than very moderate proportions.

The use of the induced-draft fan which accompanied the introduction of economizers made it possible to remove the gases from the furnace as fast as the coal could be burned, and operating men began to speculate and experiment with the possibilities of operating at very much higher ratings. It was found perfectly possible as early as 1917 to operate the 11,400-sq. ft. boilers in the River Station of the Buffalo General Electric Company at ratings nearly twice as high as those up to that time considered possible. These boilers were equipped with double-ended underfeed stokers. As attempts were made to operate boilers at higher ratings, it was found advisable to increase the furnace volume per pound of coal burned in order that the gases would be thoroughly burned and that the particles of coal blown up from the fuel bed would be well burned out before entering the first pass of the boiler. It was further noted that an increase in furnace volume decreased the amount of clinker trouble in the fuel bed. Increasing the heights of the furnaces and the size of the walls which looked in on the furnace introduced many difficult problems which had not before been encountered, these problems relating to the expansion of the walls and the ability of firebrick at high temperatures to take compressive loading. The magnitude of some of these problems can be visualized by comparing the 1300-cu. ft. furnace referred to in the typical station of 1913 design with the 21,000-cu. ft. furnace having a total width of 30 ft., a depth of 19 ft., and a total height of approximately 35 ft.; this being the description of one of the furnaces being installed in the Gould Street Station in Baltimore recently placed in operation.

We do not as yet know just how far we can safely go in increasing the output from boilers and furnaces. We do know that in the main the limitations are in the furnace and have to do with the formation of clinkers in the fuel bed, the formation of slag on the lower row of boiler tubes, and with brickwork maintenance. As pointed out later, new types of boilers embodying new problems with regard to circulation will be used. Conceivably unforeseen troubles may develop in these newer types of boilers and furnaces, and we must feel our way carefully in attempting to develop higher capacities in individual boiler units.

The use of preheated air and the desire to limit the total amount of air required for burning the coal to approximately 125 per cent of that theoretically required, indicated a further increase in furnace temperatures and further troubles with firebrick walls.

As the first step in correcting these troubles, walls of ventilated construction were used, the air used for burning the coal being passed through the hollow walls before entering the furnace. It was possible to effect marked improvements in operation by the use of this construction. Ventilated walls were, however, of complicated design, and there was some doubt as to their standing up in connection with preheated-air temperatures of from 400 to 500 deg. fahr. The logical solution seemed, therefore, to lie along the lines of eliminating to as great an extent as possible firebrick furnace walls, and in order to do this it is necessary to build the boiler around the furnace and let the boiler itself form the walls of the furnace. This is precisely what is being done in most new stations. The boiler tubes themselves form the wall. In another design the boiler tubes back up the cast-iron blocks which form the walls. The boiler tubes connected directly into the boiler circulation form a part of the boiler. Furnaces of the water-cooled wall design, although quite expensive, are well justified for the following reasons:

1 With the removal of slagging and fusing of the brickwork in the furnace walls as a limiting factor, it is possible in furnaces having water-cooled walls to burn appreciably more coal per cubic foot of furnace volume and to develop correspondingly increased capacities from a given boiler. This results in a reduction in fixed charges per unit of station output.

2 They permit the burning of the coal with a smaller amount of excess air and a resulting improvement in the overall boiler and furnace efficiency.

3 The furnace maintenance is reduced and the number of hours per year that the boiler is available for service is increased. This in turn reduces the investment in spare-boiler capacity.

Improved Methods of Burning Coal

An outstanding development of the last ten years in power-station design has been the introduction of pulverized-fuel firing into large steam generating stations. In 1917 a pioneer installation of ten boilers was made in the station of the Puget Sound Traction, Light, and Power Company in Seattle. The first major installation was made in the 40,000-kw. Lakeside Station in Milwaukee in 1921. This was followed in 1923 by the construction of the Cahokia Station in St. Louis and the addition to the Lake Shore Station of the Cleveland Electric Illuminating Company of Cleveland. In the Lake Shore Station at Cleveland it was demonstrated that test efficiencies of as high as 92.9 per cent could be obtained and that an overall gross boiler-room efficiency of 90.4 per cent could be obtained for a month's time, this in connection with a combined installation of boilers and economizers. An even more striking achievement of a gross boiler-room efficiency of 81.2 per cent over a month's time has been obtained in the Cahokia Station without the use of economizers and with an inferior grade of southern Illinois coal. An achievement of equal interest with pulverized-fuel firing is that of carrying a sustained load of 13,000 boiler hp. or 400,000 lb. of water per hour evaporated into steam for a period of as long as five days at a time, on one of the 26,470-sq. ft. boilers for the River Rouge Plant of the Ford Motor Company.

A great many installations of unit pulverizers have been made in the last two years, and there are many engineers who believe that the real future of pulverized-fuel firing lies along the lines of installing one or more mills in connection with each boiler unit, these mills taking coal from the overhead bunker and delivering it direct to the burners in pulverized form. Most of the pulverized-fuel operating experience, in connection with which careful operating records have been kept, has been in stations using central pulverizing plants. We do not at the present time have sufficient evidence upon which to form a final judgment as to the operating results which can be obtained in connection with pulverized-fuel systems using the unit mills.

Pulverized-fuel firing has been a real contribution to the art of power-station design and operation of itself, and also because it has pointed the way to major improvements in furnace design for stoker installations.

Running in parallel with the developments in pulverized-fuel firing have been major improvements in the design of stokers. The underfeed stoker has been developed in sizes 30 ft. wide and 20 ft. long. Chain-grate stokers of increased length and width are available. These are now being installed with front and rear arches which effectively reduce the amount of combustible going over the rear end of the stoker. These arches are also effective in mixing the rich gases from the front end of the stoker with the lean gases from the rear end of the stoker. Underfeed stokers are available for operation with air temperatures of 400 deg. fahr. and for the burning of as much as 35,000 lb. of coal per hour in a single furnace.

The developments in the design of stokers and related furnaces have been such as to require a very thoroughgoing analysis of the comparative designs and of all the operating conditions to indicate, for any particular station, whether the lowest costs of power on the station bus bars can be obtained with the installation of stokers or pulverized-fuel-burning equipment.

Auxiliary Drive

The extensive use of the regenerative cycle for feedwater heating which tended not only to displace the economizer but also the use of turbine-driven auxiliaries, led to the almost universal use of motor drive for the station auxiliaries, with the rather surprising result that the unaccounted-for losses, which in many older stations amount to from 10 to 12 per cent, dropped to a value of less than 5 per cent.

There is a tendency toward a slight increase in the amount of auxiliary power used in steam generating stations. In many cases it is possible to improve the performance of equipment materially by backing it up with a greater amount of auxiliary power, and considering the cheapness of the power generated in the newer stations, this often proves to be the easiest way of accomplishing

the desired result. Examples are the increase in heat-transfer rate for surface condensers with increased water velocities through the tubes; increased heat transfer in air preheaters and economizers, with higher gas velocities over the surfaces of this equipment; improved cycle efficiency due in part to the use of increased power for the boiler-feed pumps; and improved boiler-room efficiency incident to the fine pulverization of coal. The trend toward the operation of boilers at higher ratings and the use of air preheaters has resulted in an increase in the amount of power required for forced- and induced-draft fans in the boiler room. It is well within the limits of good practice for the total amount of auxiliary power to be from 5 to $6^1/_2$ per cent of the total power generated by the main units. These trends toward an increase in the percentage of auxiliary power taken together with the marked increase in the size of the individual units in the boiler room and turbine room have resulted in the present-day use of motors for driving many of the auxiliaries which compare very well in size with the generators which were being installed in power plants built 25 years ago.

Miscellaneous Improvements

Time is not available to tell of all the other detail improvements which have accompanied the major trends already discussed. Mention may be made of the use of evaporators for handling the make-up to the boilers; the use of the closed system of ventilation for the generators, in which the same air is used over and over again and is cooled by radiator fin-tube coolers, its heat being absorbed by the condensate going to the boilers; the use of surface condensers of improved design in connection with which a large percentage of the surface is active at all times; the use of automatic combustion-control equipment; the development of highly efficient steam-jet air pumps; the use of larger and more efficient mills for pulverizing coal; and the use of steam and waste-heat driers for removing the moisture from the coal before it is pulverized.

These developments, with certain others resulting from studies as to how the power supply for motor-driven auxiliaries can best be safeguarded, have helped to round out the development of steam-station design and enable us to build stations today that are very much more efficient and also more reliable and easier to operate than the stations designed in 1913.

Increases in the Size of Equipment

Certain developments of an entirely different character were in the main forced upon power-station engineers by the rapid growth in load and by the interconnection between the properties of the various electric light and power companies. Supplementing these underlying reasons was the desire to reduce fixed charges on investment and the labor costs.

Since 1913 the growth in size of units is shown by the following installations which were placed in operation in the years indicated:

In 1914, the first 20,000-kw. single-cylinder turbine; in 1915, three 30,000-kw. turbines of the cross-compound type and one 30,000-kw. turbine of the tandem-compound type; in 1916, a 35,000-kw. turbine of the tandem-compound type and a 35,000-kw. single-cylinder turbine; in 1918, a 45,000-kw. single-cylinder turbine and a 45,000-kw. turbine of the cross-compound type; in 1919, a 60,000-kw. three-cylinder cross-compound turbine; and in 1926, a 60,000-kw. single-cylinder turbine, a 77,000-kw. cross-compound turbine, and an 80,000-kw. cross-compound turbine. Orders have been placed within recent months for a 94,000-kw. tandem-compound turbine and for three cross-compound turbines of 160,000 kw., 165,000 kw., and 200,000 kw., respectively.

In 1913 but few boilers were in operation having areas of larger than 10,000 sq. ft. each, although it is true that the Detroit Edison Company, taking the lead, had already installed a number of boilers of the double-ended Stirling type having a total area of 23,650 sq. ft. each. It was not until 1921 that boilers of as large as 20,900 sq. ft. of the Babcock & Wilcox type were installed in the Colfax Station at Pittsburgh. Since that time there have been installed double-ended Stirling boilers having a total area of 30,000 sq. ft. each, and Babcock & Wilcox boilers having an area of 26,400 sq. ft. each. There have recently been placed in operation in a central-station heating plant of the Detroit Edison Company several double-ended Stirling boilers having an area of 41,500 sq. ft. each, and there have been ordered for the Long Beach Station of the Southern California Edison Company, Babcock & Wilcox boilers having a total area of 34,162 sq. ft. each.

The increase in the maximum kilowatt output from an individual boiler has been more than would be indicated by the increase in the area of the boilers. It was a rare thing to develop more than 4000 kw. from a single boiler unit in 1913. At the present time, however, a steam output corresponding to the generation of 45,000 kw. has been developed with one boiler now in operation, and two other boilers recently placed in operation have been designed to carry a maximum load somewhat in excess of 45,000 kw. per boiler.

The increases in the size of equipment and the installation of centralized and semi-automatic combustion-control equipment have been responsible for sweeping reductions in the amount of operating labor required. Actually less physical exertion is required today in the operation of the boiler which carries a load of 45,000 kw. than was required back in 1913 with the boiler carrying a load of 4000 kw.

Relative Values of Factors Responsible for Reduction in Coal Consumption

It is interesting to form a judgment as to the relative effects which the different factors have had in effecting the reduction in coal consumption from 1.6 lb. per kw-hr. to 1.0 lb. per kw-hr., the value which it is thought may be taken as typical of the best average practice in our newer stations. In Table 1 the author has given his judgment as to the relative weights of the different factors which have been responsible for the sweeping reduction in coal consumption. It is to be borne in mind that this analysis is based on certain generalized approximations, and it is of value only as it enables us to fix the relative importance of the different improvements in our minds.

Mercury-Vapor Turbine Installation, South Meadow Station, Hartford, Conn.

One of the most interesting power-station installations now in the course of construction is an addition to the South Meadow Station of the Hartford Electric Light Company. Following up the experimental work done in the Dutch Point Station, the Hartford Electric Light Company is now installing a mercury-vapor turbine which will have a capacity approximately five times as large as any previous installation of the kind. This unit, now being installed in the South Meadow Station, will be watched with interest, for it is of such size as to clearly indicate the adaptability of the mercury-vapor cycle for use in large steam generating stations.

Future Possibilities

Taking a look into the future, it seems safe to predict that we shall see—

1 The use of steam at still higher temperatures. Considerable operating experience has been gained during the last two or three years with steam temperatures ranging from 750 to 900 deg. fahr., most of this experience being unpremeditated and of short duration and taking place in connection with the initial operation of some of the large steam generating stations. This experience has indicated that higher steam temperatures may be used with only a reasonable amount of trouble. Manufacturers are ready to furnish superheaters, valves, and turbines designed for 900 deg. fahr. total steam temperature. There is a doubt, however, as to whether the metals which are available today will withstand the combined effects of the stresses incident to high steam pressure and a temperature of 900 deg. fahr. for extended periods of time without occasional failures.

2 An increase in the use of pulverized fuel. The trend of pulverized-fuel firing will, the author believes, be along the lines of using burners which will give a turbulent mixing of the air and coal streams, resulting in a reduction in the size of the furnaces. In order that pulverized-fuel firing shall be increasingly attractive, it is necessary that the incremental investment incident to this method of firing, as compared to the use of stokers, must be reduced, and for this reason it is probable that there will be some worth-while developments in the use of unit pulverizing mills.

3 Further increases in the size of turbine units and boilers. There

will be a tendency, as the loads on large interconnected systems grow higher and higher, to use turbine units of sizes ranging from 10 to 20 per cent of the total system peak load. A careful balance struck between the reduction in investment per unit of capacity, the improved performance, and the reduced labor costs on the one hand, and the increased investment incident to the additional spare capacity made necessary by use of the larger units on the other, will serve to limit the size of turbine units. The more attractive possibilities lie along the lines of using larger units in the boiler house as there are greater potential savings in investment incident to the use of larger units in the boiler house than in the turbine room.

4 In the main, the author believes the most attractive possibilities in steam-station design lie for the next few years along the lines of a consolidation of the gains which have already been made, the simplification wherever possible of new designs, and the adaptation of certain tendencies in design which will permit reductions in investment with but slight increases in coal consumption per unit of station output.

TABLE 1 RELATIVE VALUES OF DIFFERENT FACTORS EFFECTIVE IN REDUCING COAL CONSUMPTION FROM 1.6 LB. PER KW-HR. TO 1 LB. PER KW-HR.

	Coal consumption per kilowatt-hour of net send-out for a typical station placed in operation for the year 1913, lb		1.6
	Reduction in coal consumption in pounds per kilowatt-hour resulting from		
1	Increase in steam pressure	0.097	
2	Increase in steam temperature	0.087	
3	Use of higher boilers and air preheaters or economizers	0.11	
4	Use of regenerative cycle for feedwater heating	0.067	
5	Elimination of stray losses due to the use of electric drive for power-station auxiliaries	0.084	
6	Improvement in the performance of surface condensers and their auxiliaries	0.025	
7	Use of automatic combustion-control equipment	0.017	
8	Gain due to reclaiming in the condensate the losses from the generators and turbine oil coolers	0.008	
9	Improved performance of turbo-generators (increase in overall Rankine-cycle efficiency ratio)	0.05	
10	Improvement in combustion efficiency due to improved fuel-burning equipment and improved furnace design	0.055	
	Total reduction in coal burned per kilowatt-hour of net station send-out, lb		0.6
	Coal consumption per kilowatt-hour of net send-out for a typical station placed in operation for the year 1926, lb		1.0

Discussion

JOSEPH G. WORKER[1] wrote that the author had given a very concise and accurate review of what had happened in steam-station design in recent years. Of course there could be no differences in opinion if the facts were recorded properly, and there could be no question that engineers who had built steam generating stations had gone through improvements that, to say the least, had been startling.

Mr. Worker was particularly interested in the author's prediction for the future, especially in connection with fuel burning. There was no question but what our fuel-burning methods would change rapidly within the next ten years. The author's look into the future predicted the trend of pulverized-fuel firing along lines of unit pulverizers. There seemed to be a cautious attitude taken by manufacturing capital with respect to undertaking to develop and perfect this process of coal pulverization, and this was possibly due to the uncertainty of future market for such a device. At the present time, work was progressing with the application of, at least, partial gas recovery in connection with underfeed-stoker elements burning the residue.

The author had spoken of the rapid development of underfeed stokers and it was interesting to know that stokers were being laid down now that would feed coal through retorts 26 ft. long. These stokers were built practically like a machine, and would burn 62,000 lb. of coal per hour.

The author also mentioned the use of preheated air in connection with stokers. Underfeed stokers at the Chester Station of the Philadelphia Electric Company had operated for some time with air preheated to 500 deg. fahr., and at times had actually operated with air at 700 deg. fahr.

E. H. Tenney[2] wrote that the paper was of especial interest at the present time for the reason that the rapid improvement in station economy since 1923 had begun to slow down, and now was a fitting time to take stock, as it were, and review the important and rapid developments of the past few years. The paper did this admirably.

Of course there were developments still under way which promised additional economy and further reductions in investment and operating costs. One of these was the development of the air preheater. As explained by the author, the advantages of "bled-steam" feedwater heating had to some extent supplanted the economizer. Consequently the air preheater was being developed to take its place. In most cases air preheaters required fans, with the result that forced draft was now economically available where it had not been available before—especially in pulverized-fuel plants. This would tend to simplify the problem of establishing a means of developing the very necessary and desirable turbulence in powdered-coal burners. Preheated secondary air passing through burner deflecting vanes imparted a whirling motion to the fuel and resulted in a shorter flame length and earlier complete combustion. These two developments, as pointed out by the author, were thus proving to be supplementary to each other. This arrangement was to be used in the third section of the Cahokia Station in St. Louis, where unit mills would supply fuel through horizontal burners equipped with preheated air under pressure.

The table presenting a summary of the relative values of different factors which had brought about the reduction in coal consumption was of great interest. This was particularly true of item 10, which stated that the improvement in combustion efficiency was responsible for 0.055 of the total 0.6 lb. reduction in coal required per kilowatt-hour. This 0.055 lb. of 14,200-B.t.u. coal for improvement in combustion efficiency represented but 3.6 per cent of the 12 per cent improvement in boiler-efficiency which the author allowed. He did not say how he had arrived at his division, but a cursory examination of boiler-efficiency records would appear to indicate that a greater proportion than 3.6 per cent of the 12 per cent improvement was due to combustion improvements. It would appear that this did not give full credit to the improvements in stoker design or to the development of pulverized fuel. It would be of great interest if the author would explain how he had arrived at his figures.

Another item that had undoubtedly accounted for a part of the improvement, and one not mentioned in the tabulation, was the development of water walls and screens, and the addition of other radiant-heat surfaces in the furnace. This was no negligible item, and cases where it had accounted for at least 3.0 per cent in boiler efficiency had been reported.

Richard H. Morris[3] wrote that in view of the widespread interest that the paper had created and the importance of high temperatures, it might be of interest to many if he would add a few words regarding the relative importance of metal strength and steam dissociation when placing 900 deg. fahr. as a probable maximum temperature, and also the possibility of metals being developed which would allow this value to be exceeded.

Mellanby and Kerr had shown, theoretically at least, based on the creep limit of the metals, that chrome-nickel steel for radiant-heat surfaces gave the same temperature safety margin at 1500 lb. as 0.35 per cent carbon steel did at 250 lb., and that for convection superheater surfaces this same safety margin of 400 deg. fahr. was also given by chrome-nickel steel for 900 deg. fahr. temperature.

This offered as limits on one side 1500 lb. and 900 deg. fahr., which agreed closely with the limit of about 900 deg. fahr. imposed by the dissociation of steam, that was, the liberation of free hydrogen and oxidation of the metal. As Mr. Morris understood it, this dissociation was within reasonable limits up to 900 deg. fahr., but increased rapidly as the temperature was raised.

If it was found that this dissociation took place with all metals in the same degree and at the same temperature, it might place a definite limit which no amount of metal research and development could erase. If, however, special methods could be developed to delay this action or resist the corrosion, it was likely that strength requirement would again be the sole limiting factor. Again, the effects of dissociation might be so small that from a practical standpoint they might be neglected.

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Pumping Clay Slurry Through a Four-Inch Pipe

Results of Experiments in Pumping Clay Slurries Containing Various Percentages of Solids—Friction Losses When Pumping Through Pipe—Best Velocity for Pumping—Viscosity of Slurries

By W. B. GREGORY,[1] NEW ORLEANS, LA.

THIS paper gives the results of experiments in pumping a clay slurry, and the friction losses when pumping the material through a 4-in. cast-iron pipe.

The experiments were undertaken to help solve a practical problem for the Louisiana Portland Cement Company. This company is building a cement factory on the Industrial Canal in New Orleans. One possible source of raw material for making cement is the filtration plant of the New Orleans Water Works. Water is pumped from the Mississippi River into settling basins where the heavier particles of sand and grit are accumulated. The water is next treated, receiving a certain amount of lime and iron, depending on its source and condition of turbidity. After flowing for about a mile through concrete passages, it is again placed in settling basins. The mud which accumulates in the bottom of these latter basins is found to be excellent material for making portland cement. Sometimes it is of such chemical consistency that it needs no additional materials, but usually it is deficient in lime, which may be added in the form of ground shells.

It is about 45,500 ft. from the filtration plant to the cement factory, and if it is finally decided to use this material, about 60,000 tons per year of dry material must be transported from one plant to the other.

It is not the author's purpose to discuss the entire problem involved, as this would necessitate going into the question of ridding the slurry of water at one plant or the other, the economic limits of quantity of water in the slurry, etc.

The material accumulated in the grit reservoirs and the clay collected in the second set of basins were used for several years to fill vacant property belonging to the Sewerage and Water Board, but is now pumped to the Mississippi River through an 18-in. main.

Layout of Experimental Plant

On the 18-in. discharge main just referred to, a saddle was placed and a 4-in. cast-iron pipe line was run down General Ogden Street past the filtration plant to a vacant field where the mud could be discharged without inconvenience. The total length of pipe was 369.75 ft. Gate valves placed near both ends of this pipe gave perfect control of the flow up to the pressure available in the 18-in. main.

Near the outer end of the 4-in. pipe there was a straight run of more than 200 ft., so that it was possible to obtain the difference in pressure in exactly 200 ft. of pipe.

By means of elbows the pipe was conducted vertically, and at the top a short length of pipe attached to the riser by means of a loose elbow enabled the discharge to be shifted to either of two tanks, 5 ft. in diameter and about 6 ft. high, in which the volume of discharge was measured.

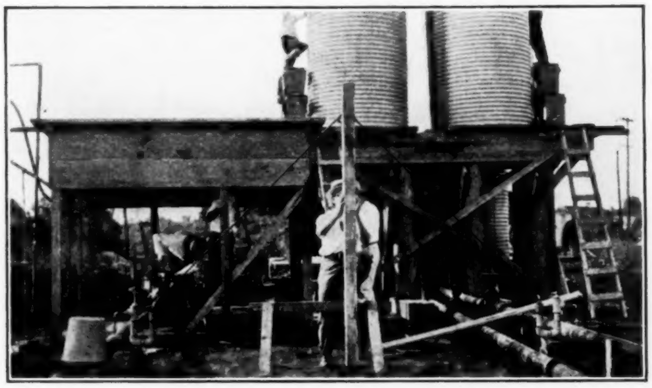
The mean areas of these tanks were found to be almost exactly the same, namely, 19.44 sq. ft. Large discharge valves near the bottom of the tanks enabled them to be emptied quickly. The 4-in. pipe was calipered before it was laid and found to be practically 4 in. in internal diameter.

A measuring rod was provided that was divided into feet, tenths, and hundredths. The velocity in the 4-in. pipe was calculated from the volume in the measuring tanks by means of the formula

$$V = \frac{19.44 \times h'}{\text{seconds} \times 0.0872} = \frac{223\, h'}{\text{seconds}}$$

where h' is the rise in feet during the observed time in seconds, and 0.0872 is the area of the 4-in. pipe in square feet.

The pressure drop in 200 ft. of 4-in. pipe was measured by means of a mercury differential gage. Openings were made on top of the 4-in. pipe by drilling holes for $^1/_4$-in. nipples, and $^1/_4$-in. pipe was used to connect to a mud trap made up of 2-in. pipe fittings. A union was provided for quickly removing and cleaning the mud traps, and a stop valve was provided for shutting off the pressure from the 4-in. pipe. From the top of the mud trap a blow-off valve was used to rid the connection of air, and from the other leg of the mud trap a $^1/_4$-in. connection led to the mercury differential gage. Before each run the mud traps were removed, thoroughly washed, and filled with water. Before observations were taken, any air that had accumulated was blown out of the mud traps and from the top or both sides of the mercury differential gage. The arrangement of mud traps is clearly shown in Figs. 1 and 2, although the complete set-up as shown in these figures is to be described later.



Figs. 1 and 2 Arrangement of Mud Traps

A few measurements were taken with the set-up described above, and it was found that with clear water flowing through the pipe the losses were practically identical with those given in Williams and Hazen's Hydraulic Tables for the very best clean, new cast-iron pipe; the designation in their tables is in the column marked (00). These results are not given for the reason that the measurements were repeated with the later and more elaborate set-up, and results were obtained that were practically identical.

The experiments described above had to be made when the mud pump at the filtration plant was in operation, and the consistency of the material pumped was found to be quite variable. The head available to cause flow was also limited, and when the valves were wide open velocities below 7 ft. per sec. were the highest obtainable. It was therefore decided that a new set-up was needed, in which the conditions of the experiment could be better controlled.

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Final Layout of Experiment

The 4-in. pipe from the discharge of the mud pump of the filtration plant was left as a supply pipe to a tank 12 ft. in diameter and about five feet high. This tank could be filled at any time the mud pump was operating. It was provided with a discharge pipe which extended inside the tank, and had a swiveled elbow and pipe within the tank so that water collecting on the surface of the slurry could be skimmed off and allowed to flow from the tank.

To the bottom of the 12-ft. tank was flanged the suction pipe leading to the centrifugal pump. The suction pipe was 6 in. in diameter and was provided with a 45-deg. elbow and a gate valve. Beyond the gate valve was a cone reducer to 4 in. diameter at the suction of the pump.

The pump used was built by the Allis-Chalmers Company for

Fig. 3 Layout of Experiment

valve flanged to the 4-in. pipe. Pressures were read at the pump by means of two calibrated bourdon pressure gages reading in feet of water. The gage on the suction pipe was connected to the middle of the reducer where the diameter was 5 in. The gage on the discharge was at the middle of the expander where the diameter was $3^1/_2$ in.

The 4-in. cast-iron pipe extended out from the pump a distance of approximately 116 ft. to a 90-deg. flanged ell having a 9-in. radius, then across to another 90-deg. flanged ell having a radius of $6^1/_2$ in., and then back to the measuring tanks. The layout is plainly shown in Figs. 3 and 4. The distance from center to center of the long pipes was 13 ft. $3^1/_2$ in.

The arrangement was such that it was possible to tap the 4-in. pipe at points 100 ft. apart, and at a distance from the stop valve of the pump and the ells at the loop of about 8 ft. or 24 diameters. Then by connecting the two openings near the pump and the measuring tanks to a common mercury differential gage, the loss between these two points could be observed. By connecting the two openings near the loop to another mercury differential gage, the loss in the loop consisting of loss in straight pipe and elbows could be determined. The difference between these two sets of readings gave the loss in 200 ft. of straight pipe.

The measuring tanks were elevated to discharge into the large suction tank, so that material was not wasted but was pumped over and over through the system of pipes and tanks.

The Mercury Differential Gages

The differential gages are plainly shown in Fig. 2. They have two glass columns side by side in an aluminum channel with iron fittings at top and bottom. Behind the glass tubes is an adjustable scale, graduated in inches and tenths, so constructed that it may be raised or lowered until the zero is at the height of the mercury column.

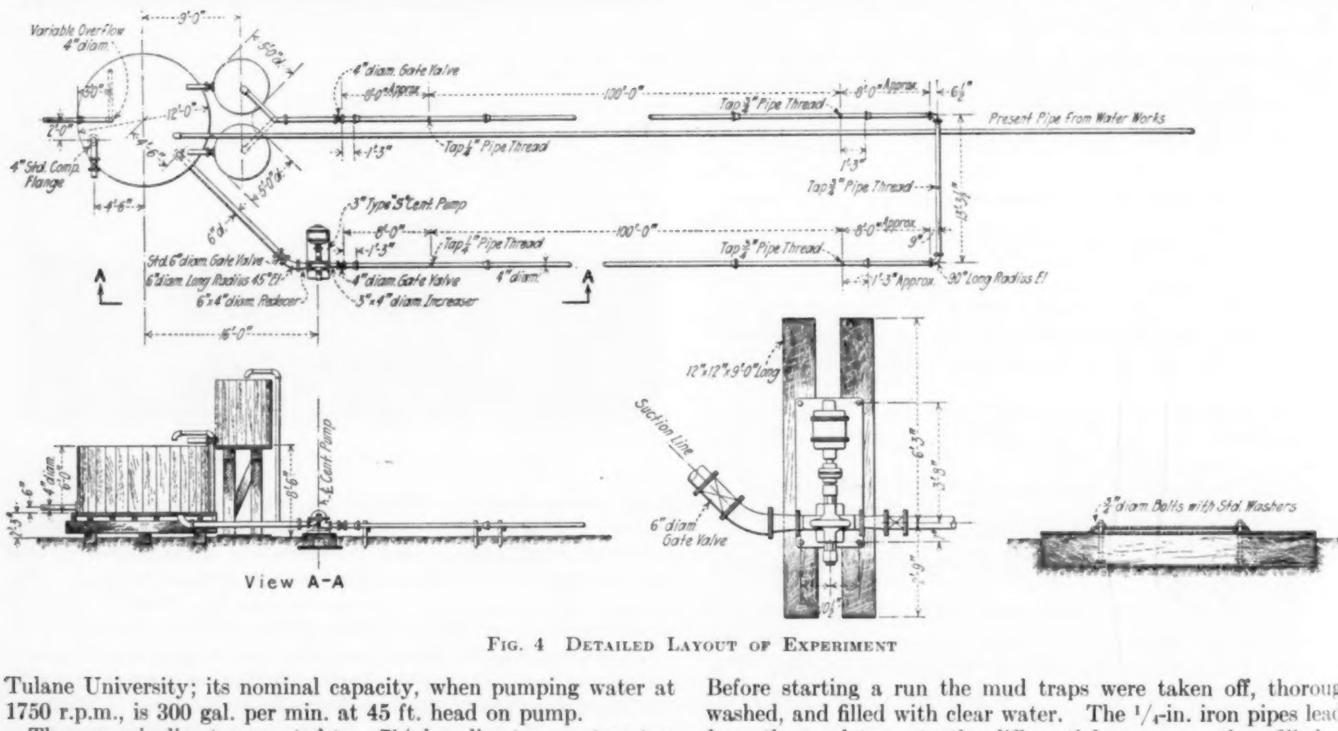


Fig. 4 Detailed Layout of Experiment

Tulane University; its nominal capacity, when pumping water at 1750 r.p.m., is 300 gal. per min. at 45 ft. head on pump.

The pump is direct-connected to a $7^1/_2$-hp. direct-current motor, and speed control is obtained by means of two rheostats. While the normal voltage is 115, it was found desirable to furnish a voltage of 220 on account of better speed control, as the variation of voltage was considerable due to external causes that could not be eliminated. Electrical energy was furnished by means of a cable from the filtration plant.

The suction of the pump was 4 in. in diameter and the discharge 3 in. The suction line from the tank was 6 in. in diameter. A reducer from 6 in. to 4 in. was used at the pump; back of the reducer was a 45-deg. elbow and beyond that a 6-in. gate valve. During these tests this valve was always wide open. Beyond the discharge flange a 3-in. by 4-in. expander was used, followed by a 4-in. gate

Before starting a run the mud traps were taken off, thoroughly washed, and filled with clear water. The $^1/_4$-in. iron pipes leading from the mud traps to the differential gage were then filled with water and connected, while the glass tubes above the mercury had previously been filled with water. Care was taken to remove air pockets, or, in other words, that the water columns were completely filled with water. The differences of height were read in inches and tenths of inches and reduced to feet of water by multiplying inches difference by 1.05. Since the specific gravity of mercury is 13.6 and that of water is unity, we have:

$$h = \frac{z(13.6 - 1)}{12} = z(1.05)$$

where h = head in feet of water and z = reading of differential gage in inches.

Scope of the Experiments

While the primary reason for conducting the experiments was to study the losses in pipes when pumping clay slurry, it was thought probable that the pumping, if done at all, would be accomplished by means of centrifugal pumps; and inasmuch as such a pump was used in the final experiment, it was thought worth while to conduct a test of the pumping plant. While some of the results of the pump test were unsatisfactory, the results in general were of interest and of some value.

The results obtained from the experiments on the loss of head in pipes are, as far as the author is aware, the only data available on this problem.

Methods of Conducting Tests

The plan adopted in conducting the tests of May, June, and July was in general to take three sets of readings with conditions as constant as possible and to average the results. In the tests of November and December there were sufficient observers to take all observations simultaneously, and single observations were plotted. In general, the revolutions per minute were kept between the limits of 1600 and 1650, with an attempt to approximate 1625.

When conditions were satisfactory, observations were taken of time, r.p.m., amperes, volts, suction and discharge gages, the drop in pressure in 200 ft. of 4-in. pipe plus an additional length of 29.39 ft. of straight pipe and two ells, one having a radius of 9 in. and the other a radius of 6 in., and the drop in pressure in the loop of shorter pipes and elbows. Temperatures of air and of material pumped were read from time to time.

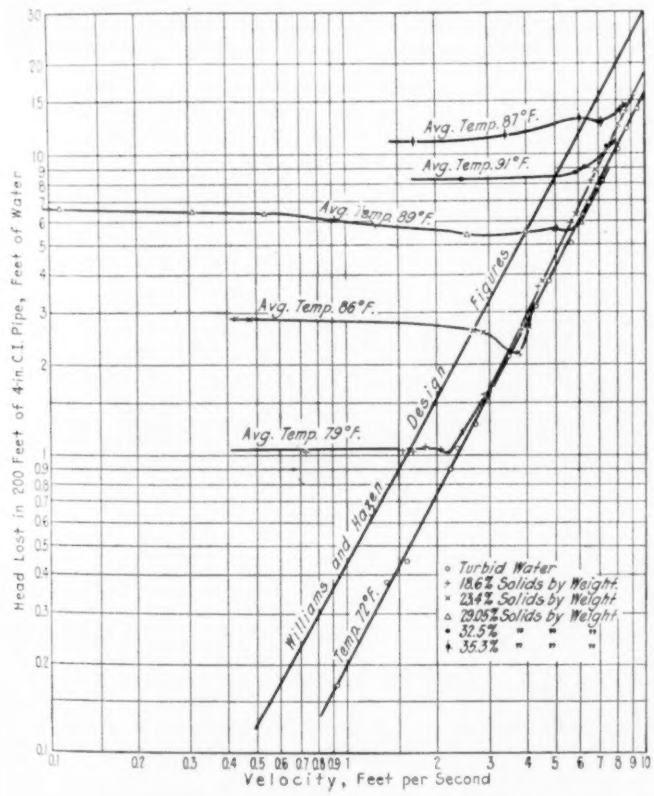


Fig. 5 Results of May and June Experiments

The three observations for a given set of conditions were usually taken at about five-minute intervals, and it was found that it required about that time to make the rounds and record the readings.

Meanwhile readings of quantity pumped were obtained in the smaller tanks. When a measurement of rise of level in the smaller tank had been completed and the time for the rise had been observed by means of a stop watch, the material was allowed to flow into the larger tank by opening the drain valves. A swinging pipe allowed the discharge to be turned into either of the smaller tanks. The velocity in the 4-in. pipe was obtained from the quantity-time observation as already explained.

At the end of a set of three observations the average velocity and the average loss in head in 200 ft. of pipe were computed. The results were then plotted on logarithmic paper so that any unusual conditions and the progress of the experiment could be noted.

The setting of the gate valves beyond the pump was then changed, the speed adjusted, and when conditions were constant and satisfactory another set of readings were obtained.

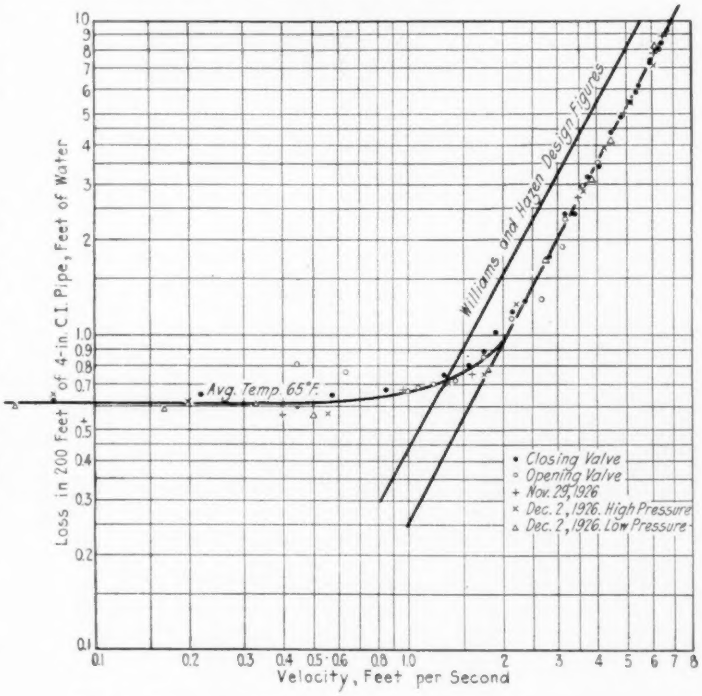


Fig. 6 Results of November and December Experiments

Before starting the observations when clay slurry was pumped, a run was made pumping clear water. Perhaps it would be more exact to say turbid water, as it was obtained from the settling basins but it was practically clear water. This was desirable as the losses when pumping water gave a standard with which the other results could be compared.

When this work had been completed the work on the clay slurry was begun. The record is as follows:

May 12–14, 1926. Made experiments with turbid water.
May 17, 1926. Material placed in experimental tanks.
May 20, 1926. First experiments with 18.6 per cent solids by weight.
May 22, 1926. Made experiments with 23.4 per cent solids by weight.

The slurry was then allowed to settle and surface water was drawn off. Some of the slurry was placed in an earthen reservoir where the water seeped into the soil, after which the slurry was returned to the experimental tanks.

June 1, 1926. Made experiments with 29.05 per cent solids by weight.
June 18, 1926. Made experiments with 32.5 per cent solids by weight.
July 8, 1926. Made experiments with 35.3 per cent solids by weight.

The results are shown in Fig. 5.

The later experiments of November and December were undertaken to settle more definitely the direction of the line for points below the critical velocity. The first set of tests had shown that losses below the critical velocity were practically independent of the velocity. To settle this point a new sample of slurry was obtained from the settling basins and experiments run as shown in Fig. 6.

The loss of head was read with a mercury differential gage as described above, but the low readings were from gages having water columns with air above the water, so that losses were read directly in feet of water. Samples of data as observed and computed are tabulated at the top of the following page.

The Material Pumped

The material pumped does not differ greatly from 15 per cent solids by weight as it comes from the reservoirs of the filtration

plant. By allowing it to settle for several days and drawing the water off the top of the mixture, it may be concentrated to about 25 per cent solids by weight. To further remove the water by settling is a slow process, requiring considerable time.

PUMP TEST—CLAY SLURRY (23.4 PER CENT SOLIDS)

Time	R.p.m.	Amp.	Volts	Watts	Elec. hp.	Gages—Suction as read	Gages—Suction corr.	Gages—Discharge as read	Gages—Discharge corr.	Head on pump, ft.	V	Q, G.p.m.	Pump hp.	Effy., Per cent
10:00	1610	17.2	126.0	2170	2.90	3.2	3.9	55.7	57.3	53.4	0	0	0	0
10:28	1625	40.6	121.1	4930	6.60	1.0	1.7	35.9	37.4	36.9	8.12	317.0	2.95	44.6
10:50	1635	37.7	123.2	4550	6.23	2.3	3.0	39.5	40.5	38.3	6.81	266.0	2.57	41.2
11:08	1633	35.2	118.0	4150	5.56	2.1	2.8	43.2	44.5	42.0	5.82	228.0	2.41	43.3
11:25	1632	31.5	115.7	3640	4.89	2.4	3.1	48.6	50.0	47.3	4.54	177.5	2.11	43.2
1:00	1621	30.7	111.1	3410	4.57	2.9	3.6	48.3	49.5	46.2	4.06	159.0	1.85	40.4
1:20	1650	30.5	113.0	3440	4.61	2.8	3.5	50.5	51.7	48.5	3.78	148.0	1.80	39.1
1:40	1645	27.0	111.3	3010	4.04	2.8	3.5	53.7	54.9	51.6	2.89	113.0	1.47	36.3
1:57	1628	26.1	109.9	2860	3.84	2.7	3.4	53.6	55.0	51.8	2.67	104.5	1.36	35.4
2:16	1620	19.0	107.5	2040	2.74	3.0	3.7	54.9	56.2	52.5	0.48	18.8	0.25	9.1
2:33	1630	18.8	107.2	2020	2.71	3.1	3.8	54.1	55.5	47.9	0.43	16.8	0.20	7.5
3:07	1645	27.7	115.0	3180	4.27	2.7	3.4	55.7	57.5	54.3	3.15	123.0	1.68	39.4

PUMPING CLAY SLURRY (23.4 PER CENT SOLIDS)

No.	Time	R.p.m.	Amp.	Volts	Pressure, ft. of water Suct.	Pressure, ft. of water Disch.	Inches diff. gage—200 ft. plus ells	Inches diff. gage—Ells	Inches diff. gage—Diff.	Diff. in ft. of water	Discharge Rise in ft.	Discharge Sec.	Vel.	Temp.
1	10:00	1610	17.2	126.0	3.2	55.7			Stop	valve	closed			
2	10:20	1631	41.0	122.8	1.0	36.8	15.40	3.00	12.40		1.58	42.8	8.23	88+
3	10:23	1615	40.5	120.0	1.0	35.5	14.55	2.75	11.80		1.56	42.8	8.13	
4	10:28	1630	40.2	120.5	1.0	35.5	14.38	2.75	11.63		2.14	59.8	7.99	
	Avg.	1625	40.6	121.1	1.0	35.9	14.78	3.83	11.94	12.54			8.12	
5	10:43	1625	37.5	128.5	2.5	39.1	10.40	2.00	8.40		1.31	42.8	6.82	82S
6	10:47	1630	37.8	121.0	2.2	39.5	10.65	2.15	8.50		1.30	43.0	6.75	80A
7	10:50	1650	37.8	120.0	2.0	39.9	10.65	2.15	8.50		1.39	45.2	6.87	
	Avg.	1635	37.7	123.2	2.3	39.5	10.57	2.10	8.47	8.89			6.81	

LOSS IN ELLS (23.4 PER CENT SOLIDS)

I Loss in 200 ft.	II 0.147 × I, Loss in pipe	III Loss in ells and pipe	III minus II, Loss in ells	V
12.54	1.54	2.97	1.43	8.12
8.89	1.31	2.20	0.89	6.81
6.22	0.91	1.55	0.64	5.82
3.70	0.54	0.91	0.37	4.54
2.13	0.31	0.60	0.29	3.78
2.55	0.37	0.49	0.12	2.89

For the tests of May, June, and July the material tested was said to be a fair average sample, although it must be remembered that the Mississippi has many tributaries and at various times the waters from one of these sources may greatly exceed the normal flow from that stream, with corresponding changes in the nature

TABLE 1 DATA ON SLURRIES PUMPED

No.	Water Per cent by weight	Water Per cent by volume	Solids Per cent by weight	Solids Per cent by volume	Weight of mixture, lb. per cu. ft.[1]	Specific gravity of mixture	
1	48.0	71.0	52.0	29.0	92.0	1.48	Slurry pumped in May, June, and July
2	61.2	80.7	38.8	19.3	82.2	1.32	
3	69.0	85.5	31.0	14.5	77.1	1.24	
4	82.8	92.7	17.2	7.3	69.6	1.12	
5	88.2	95.2	11.8	4.8	67.2	1.08	
6	76.4	88.1	23.6	11.7	73.2	1.17	Slurry pumped in November and December

[1] Weight of 1 cu. ft. of water at 85 deg. fahr. = 62.2 lb.

of the material removed in the purification of the water. Not only is the original material itself different, but the amount of lime and iron added will affect the material that is settled out after coagulation has taken place, as well as the semi-fluid matter surrounding the solids. An analysis of the sludge or slurry pumped in the tests of May, June, and July was made by J. L. Porter, Director of the Water Purification Department, New Orleans, La., and is reported as follows:

Herewith are presented the results of analysis of the sludge from Coagulating Reservoir No. 1, at the Carrollton Water Purification Plant, as used in the investigation of pipe-line flow and pumping conditions.

The sample taken was from the large tank at a time when the total solids had reached approximately 38 per cent by weight.

	Per cent
Loss at 105 deg. cent. (water)	61.97
Loss on ignition	4.25
Silica and insoluble silicates (clay and sand)	23.40
Iron oxide and alumina	4.21
Calcium oxide	5.21
Magnesium oxide	0.76
Undetermined	0.20
	100.00
Carbonic acid (CO_2)	4.01

Calculated to the dry basis the approximate analysis would be as follows:

	Per cent
Clay and sand	70.29
Carbonate of lime	23.96
Hydrate of magnesia	2.90
Hydrated oxide of iron	2.32
Undetermined	0.53
	100.00

This material contained a small amount of sand, which was not determined, but is included in the silica residue and classed with the clay.

This shows a clay-lime ratio of approximately 3 to 1.

A sample of the wet slurry was taken on June 10, 1926, and evaporated to dryness at 212 deg. fahr. It showed 31 per cent solids by weight and 14.5 per cent by volume. Keeping the solids constant and varying the amount of water in the mixture, the values in Table 1 are obtained. Fig. 7 shows the more important relationships of Table 1.

The percentage of solids by weight is easily determined experimentally, while the percentage by volume required very careful handling. The solids present in samples Nos. 1 to 5 were found to have a specific gravity of 2.65, or a weight per cubic foot of 165 lb.

The weight per cubic foot of the slurry (No. 6 of the table) was 73.2 lb. and the specific gravity of the dry solid material was 2.38.

Under the microscope the material Nos. 1 to 5 had the appearance of minute crystals, a majority of which were of a light yellow color,

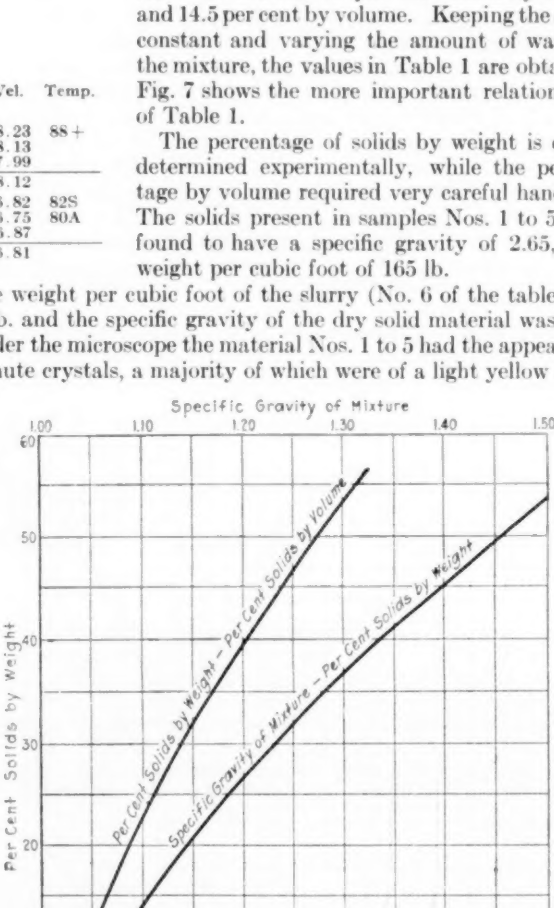


Fig. 7 Data on Slurries Pumped

with a few darker and in some cases almost black. The material used in November and December appeared under the microscope to be composed of smaller particles quite uniform in color and size.

The experiments showed that the solids did not separate easily or quickly from the water, and that they did not cake in the pipe or stick to the pipe walls. After remaining in the pipe for several days, pumping can be resumed by applying a reasonable pressure to the pipe. A long pipe line with occasional connections to city water mains and convenient blow-out valves would meet the practical requirements of commercial pumping.

Results of Experiments

Loss of Head in Pipes. Figs. 5 and 6 show the losses of head in

200 ft. of 4-in. cast-iron pipe measured in feet of water, plotted against velocity in feet per second; the first is for experiments of May, June, and July, while the second is for the tests of November and December. It will be noted that the results as shown when turbid water was flowing are on a straight line. As already stated, this line nearly agrees with the figures as given by Williams and Hazen in their Hydraulic Tables for 4-in. cast-iron pipe, clean, new, and laid perfectly straight, but the losses are slightly less.

The design figures recommended by Williams and Hazen are also given in the diagram as they do not mix with the other plotted results.

The slope of the line in the log plot for water is 1.90, and this is the value of n, the exponent of the velocity v in the formula $h = Kv^n$.

It should be noted that the experiments with water were not carried to a sufficiently low velocity to give viscous or non-turbulent flow. When the material contained solids by weight between 18.6 per cent and 35.3 per cent the log plot shows that a portion of the line substantially agreed with the line for water, indicating that turbulent flow was taking place. When the velocity was decreased to a certain value a region of critical velocity was encountered, while still further reduction of velocity shows practically constant loss of head.

It will be noted that the three lower curves in Fig. 5 follow the line for water quite closely for some distance, while for the two upper curves they start near the line for water, but soon diverge from it, and as velocity is decreased soon became nearly horizontal. Two reasons account for this peculiarity. The material pumped was more concentrated than in the previous tests and the end loop of the pipe shown in Fig. 2 was replaced by that shown in Fig. 8, the latter having greater length and adding two 90-deg. elbows and two 180-deg. elbows in addition to those of the original loop.

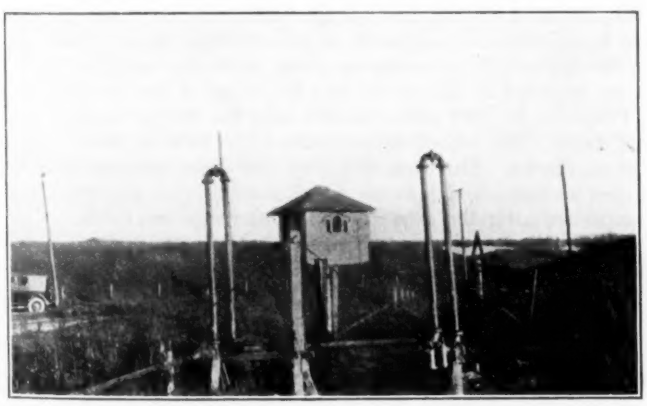


FIG. 8 SUBSTITUTED END LOOP OF PIPE

Loss of Head in Elbows. The loss of head due to two elbows, one with $6^1/_2$ in. radius and the other with 9 in. radius was determined; the results are given in Fig. 9.

The loop containing the elbows had 29.39 ft. of straight pipe, a part of which was cast iron and a part steel. Presumably the loss in this straight pipe is nearly the same per unit length as in the 200 ft. of cast-iron pipe. As the loss in 200 ft. was known from the experiments, it was assumed that the loss in 29.39 ft. would be in proportion to its length or 0.147 times that of the longer pipe.

Now the loss in an elbow is not all in the length represented by the curve itself, for a loss is caused in the straight pipe beyond the elbow, and the pressure openings of the pipe were probably far enough removed from the elbows to include the loss beyond them.

If from the actual loss observed the loss in the straight portion of pipe be subtracted, the difference should be the loss in the two elbows. As the loop was replaced after the third set of experiments with a double loop as shown in Fig. 8, no results were computed for the last four sets of readings.

THE PUMP TESTS

Pump tests are reported only for the experiments of May, June, and July.

The quantity pumped was known from the volume-time measurements in the tanks. The head on the pump was computed from the observed pressures on the suction and discharge sides of the pump corrected for velocity head.

$$H = h_d - h_s + \frac{V_d^2 - V_s^2}{2g}$$

where H = head on pump in feet of water
h_d = static head on discharge pipe
h_s = static head on suction pipe
V_d = mean velocity in discharge pipe
V_s = mean velocity in suction pipe.

As the two gages were at the same level, there was no correction for height. Both read in feet of water.

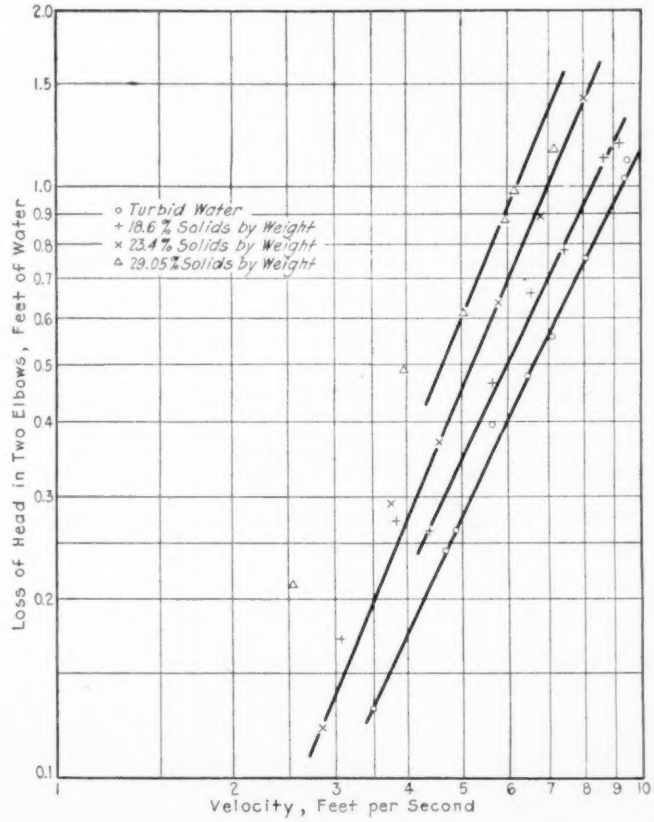


FIG. 9 LOSS OF HEAD IN TWO 4-IN. CAST-IRON ELLS
(One ell, 6 in. radius; one, 9 in.)

The horsepower required to pump a fluid through a horizontal pipe, overcoming friction losses, may be expressed as follows:

$$\text{Hp.} = \frac{WH}{550} = \frac{Qw'h\ (w/w')}{550} = \frac{avhw}{550}$$

where W = pounds per second of the fluid pumped
H = total friction head in feet of fluid pumped
h = total friction head in feet of water
w = weight per cubic feet of water
w' = weight per cubic feet of fluid pumped
Q = quantity in cubic feet per second
a = area of pipe in square feet
v = mean velocity in pipe.

It is seen that this is independent of the density or specific gravity of the material pumped, so long as h is read in feet of water and not in feet of the material pumped.

The results of the pump tests are plotted in Figs. 10, 11, and 12. It should be noted in these sets of curves that there is a sagging in all three for 29.05 per cent solids by weight. Before this test was made the material was removed to an earthen reservoir. In putting it back into the experimental tanks, some foreign matter was carried with it in spite of considerable care. Strange sounds developed in the pump during this test, and on taking it apart it was discovered that some of the passages of the pump at the entrance

to the impeller were partially closed. The obstructions were removed before the last two experiments were made—with 32.5 and 35.3 per cent solids by weight.

It was very difficult to read the suction gage on the pump as the mixture became more concentrated and less fluid. The ac-

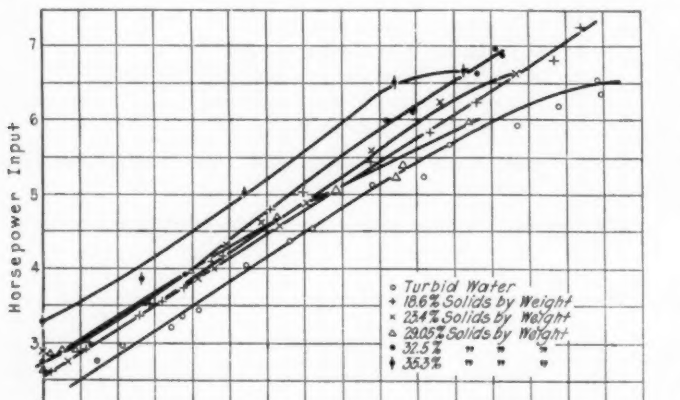


Fig. 10 Results of Pump Tests—Horsepower Input at Various Loads

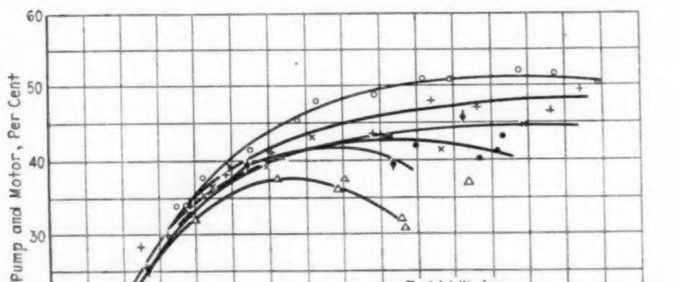


Fig. 11 Results of Pump Tests—Efficiency of Pump and Motor at Various Loads

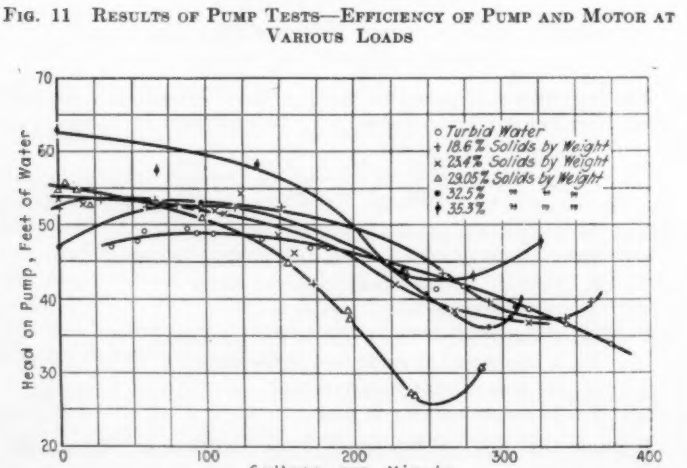


Fig. 12 Results of Pump Tests—Head on Pump at Various Loads

curacy of the last two experiments, and especially the last one, is not as great as that of the others and easily accounts for the peculiarities of the curves.

Discussion of Results

An attempt to interpret the results of these experiments led to an inquiry as to the nature of the material pumped. When the flow through the pipe was turbulent it behaved very much like water, but below the critical velocity the behavior was most unusual inasmuch as the losses due to pipe friction were independent of the velocity. It is hoped that the discussion of the paper will bring out some ideas regarding the nature of the material and the reasons for the peculiarity of the friction loss.

The substance pumped has some properties of a viscous fluid and in other ways it behaves like a very plastic solid. It is composed of minute particles of solid material surrounded by a lubricating material consisting of water and some colloidal material (clay), with very little tendency for the solids to settle out. Certainly it obeys laws of viscosity very different from those of water. But there is a critical velocity which is the economical velocity for pumping.

Viscous Flow

A fluid may flow in filaments that are parallel to the axis of the pipe, this phenomenon being known as viscous flow. When the velocity is increased so that there is considerable movement of particles from the inner surface of the pipe to the mass of fluid within, the flow is turbulent. The difference between these two sorts of flow was investigated by Osborne Reynolds, and his results were reported in the Philosophical Transactions of the Royal Society in 1883.

By using glass tubes and introducing aniline dye, Reynolds was able to demonstrate experimentally the existence of these two forms of flow and to further show that the point of change from one form to the other depended on whether the change was from a high to a low velocity or from a low to a high. In other words, he found a zone of uncertainty where the velocity was either viscous or turbulent. Experiments with water made by Reynolds and many other investigators show that when viscous flow exists the loss of head is directly proportional to the velocity. On logarithmic paper this plots as a 45-deg. line. When turbulent flow takes place, the loss is proportional to a power of velocity that is usually less than 2; the figure 1.75 is sometimes given, while the experiments with water reported in this paper give the slope of the line as 1.90.

Poiseuille, in 1842, experimented with the flow of fluids in capillary tubes. He wished to understand the flow of blood in veins and capillaries. His work was very well done, and the deduction of the mathematical laws on which they depend was quickly accomplished after the experimental results were available.

Poiseuille's law may be expressed by a formula in English units as follows:[2]

$$p = \frac{32uLV_c}{gd^2}$$

where p = pressure drop as lb. per sq. ft.
L = length of pipe in feet
V_c = average velocity of fluid at critical velocity
u = absolute viscosity in second-poundals per sq. ft. = lb-sec. per sq. ft.
g = acceleration due to gravity, ft. per sec. per sec.
d = diameter inside pipe in feet.

Viscosity is defined as the internal friction of a moving liquid. It is the force required to move a plane surface of one square area past another of large area at unit distance, at unit rate of speed, when the space between is filled with the viscous liquid. Many ways have been devised for measuring this force; as, for instance, by dragging a flat surface body through liquid and noting the pull required; however, the form of instrument most frequently used consists of a tube of small bore and relatively great length, through which the liquid is driven under a small head and at low velocity, so as to prevent eddying. Such an instrument is the Saybolt viscosimeter, used to determine the viscosity of oils. An attempt was made to determine the viscosity of the clay slurry of the experiments of May, June, and July by means of a Saybolt universal viscosimeter, but without success. With the materials of the later experiments—those of November and December, 1926—it was possible to use the viscosimeter, and the following results were obtained:

[2] Principles of Chemical Engineering, by Walker, Lewis & McAdams, p. 76.

Temperature of air and slurry, 72 to 74 (average 73), deg. fahr.
Time for 60 cc. to flow, 45 sec.

Further attempts were made to determine the viscosity of this material by Leon Lassen, working with another Saybolt standard viscosimeter, and with temperatures ranging from 43 to 126 deg. fahr. The results were somewhat discordant, but seemed to indicate that viscosity for this material was independent of temperature and the average time for 60 cc. of slurry to flow was about 48 sec.

This slurry differed from that experimented with in May, June, and July, in that it appeared to have smaller particles when seen under the microscope, and there was a great uniformity of size and color that was lacking in the earlier-tested material.

If the slurry is a viscous material, it should be possible to apply Poiseuille's law. Taking the equation that represents this law, it may be used to solve for u the absolute viscosity in these experiments at or below critical velocity.

This certainly can be done when the liquid is water or a viscous fluid like oil. With fine solid particles in suspension in a fluid there is some question as to whether it is a problem of viscosity or one of plasticity below the critical velocity. These experiments show that above the critical velocity the material behaves much like water and that there is turbulent flow.

Assuming that Poiseuille's law applies at the critical velocity, we have:

$$p = \frac{32uLV_c}{gd^2}$$

$$= \text{pressure drop in pounds per square feet}$$

$$= 0.433 \times 144\, h = 62.3h$$

$$62.3h = \frac{32u(200)(V_c)}{(32.16)(1/3)^2}$$

$$u = \frac{0.0348h}{V_c}$$

Using the log plots, Figs. 5 and 6, we may now read off values of V_c the critical velocity and h the loss of head in 200 ft. of pipe in feet of water and compute u—see Table 2.

TABLE 2 VALUES OF CRITICAL VELOCITY, LOSS OF HEAD, FRICTION FACTOR, AND VISCOSITY

No.	Per cent solid by weight	Sp. gr. of slurry, S	Critical velocity, V_c	Loss of head in 200 ft., ft. of water, h	Loss of head in feet of slurry, H	$\frac{V_c^2}{2g}$	Friction factor, f	Viscosity, u
1	18.6	1.13	2.25	1.05	0.93	0.0787	0.0197	0.0162
2	23.4	1.175	3.60	2.60	2.21	0.2015	0.0183	0.0251
3	29.05	1.225	6.00	5.70	4.65	0.5598	0.0139	0.0331
4	32.5	1.255	7.00	9.60	7.65	0.7618	0.0167	0.0477
5	35.3	1.285	8.50	14.00	10.90	1.1230	0.0162	0.0572
6	23.6	1.17	1.5	0.77	0.658	0.0350	0.0312	0.0178
7	23.6	1.17	2.0	1.00	0.855	0.0622	0.0229	0.0174

The value of the friction factor f may also be computed at the critical velocity.

$$H = \frac{fL}{d}\frac{v^2}{2g} \quad \therefore \quad f = \frac{2gHd}{Lv^2}$$

If we assume at the critical velocity both Poiseuille's law for viscous flow and the ordinary law for loss of head due to turbulent flow apply, we may write

$$p = \frac{32uLV_c}{gd^2} = \frac{fw'LV_c^2}{2gd}$$

From which we have $V_c = \dfrac{64u}{fw'd}$, where w' is the weight of fluid in pounds per cubic feet; water at 68 deg. fahr. weighs 62.3 lb. per cu. ft.

If the viscosity u, the friction factor f for the pipe in question, and the weight per cubic feet of the fluid pumped w' are known, it is possible to compute V_c for any size of pipe required. This is important from the engineer's point of view, because it is desirable to pump the slurry at about the critical velocity. If the critical velocity is greatly exceeded, the head and the losses increase nearly as the square of the velocity for turbulent flow.

Since the absolute viscosity of water at 60 deg. fahr. is 0.000672 in English units, the viscosity of the slurry experimented with varies from 24 to 85 times that of water. It should be noted that the two samples of slurry pumped showed quite different values of f the friction factor and u the viscosity. No doubt a large part of these differences was due to the nature of the material itself. The value of f was possibly increased for the tests of November and December because the pipe was several months older than in the tests of May, June, and July, and was not in as good alignment. The inner surface of the pipe had also doubtless changed somewhat due to rusting. However, all these reasons probably contribute but little to the increase of f.

The formula[3] that is considered as applicable to viscous fluids to determine the critical velocity is

$$V_c = \frac{Ku}{w'd}$$

It is seen that V_c is directly proportional to the viscosity u, and inversely proportional to the weight per cubic foot of the material w' and to the diameter of the pipe d. The effect of density of the fluid and its relationship to viscosity is brought out in the statement—a condition "tending to stability and steadiness of motion" is a "reduced density of fluid."[4] These considerations are important and should be kept in mind in what follows, as points of variance with accepted viscous-flow formulas will be shown for the material under consideration.

The results of the experiments reported in this paper as shown in the above table of results, indicate that for this material the critical velocity is greater as the density is increased, which is exactly the opposite of statement of the above formula. It is therefore questionable whether the viscosity u can be found by the methods that have been used in computing the table. The question arises, Has the slurry a viscosity comparable to homogeneous fluids, such as water and oil, for which the formula was derived? Leaving this question unsettled for the moment, let us examine the first formula for viscous flow:

$$p = \frac{32uLV}{gd^2}$$

The pressure drop is directly proportional to the velocity and to the viscosity u. Comparing this formula with the experimental results, it is seen that in the latter the loss of head below the critical velocity does not increase with the velocity V but, on the contrary, it even decreases slightly with V in some cases; in general, however, it may be said to be independent of V. The results show a failure to conform to the formula, and point to the advisability of not considering the flow of the slurry as similar to that of homogeneous liquids.

In the viscous flow of a fluid within a pipe at constant temperature it has been assumed that the velocity increases from a value of zero at the wall of the pipe to a maximum velocity at the axis; that particles of fluid travel in straight lines parallel to the axis, and that there is no radial component of the velocity. In other words, it is assumed that there is no mixing of the various concentric layers of liquid which are slipping by one another.[5] Experiments made with colored water injected into a stream of water flowing in glass tubes have demonstrated that the above assumption is substantially correct.

When the nature of the material used in the experiments of this paper is investigated, it becomes obvious that it is not truly viscous as is water or oil. The slurry, as already noted, consists of solid particles suspended in a liquid. These particles vary in size from that which is considered as colloidal to larger microscopically visible particles. If it is assumed that the flow occurs in concentric layers it must be considered that the resistance to flow is due to the shear of adjacent layers or to a displacement of solid particles within the layers. In pure viscous flow the shear is due to the viscous resistance between layers only, while in the flow of slurry the cohesion of the liquid becomes but a part of the total resistance,

[3] Principles of Chemical Engineering, Walker, Lewis & McAdams, p. 76.
[4] Hydraulics, Gibson, p. 43.
[5] The Flow of Liquids, by W. H. McAdams, Bulletin Mass. Inst. of Technology, vol. 60, no. 65.

for the layer is made up of a suspension of solid particles which interlock within and between layers, tending to increase the resistance to shear to such an extent that the cohesive resistance becomes but one factor. It is not intended to imply that the viscous resistance of the liquid is absent, but to emphasize the fact that the resistance due to the presence of the solids is of great importance.

If the viscous resistance of the liquid varies as in water, oil, and air, the losses vary directly with the velocity and it follows that the resistance due to the presence of the solids must vary in some opposite way so that the sum of the two is constant below critical velocity.

It may also be assumed that each layer contains the same number of particles at the beginning of flow as at the critical velocity, so that each layer, and hence the entire body of slurry, must have a starting resistance. This helps to explain the extraordinary fact that the slurry shows practically the same loss at extremely low velocity as with velocities that approach the zone of critical velocity.

From the above hypothesis it may be expected that with an increase of concentration of solid particles the starting resistance will increase, also that with equal concentration a suspension of a finer material will have a smaller starting resistance and hence a lower critical velocity. From the curves presented in this paper it will be seen that the above corollaries are in accordance with the experimental facts, for with a greater concentration of particles the critical velocity is increased. Also, comparing Experiment No. 2 with Nos. 6 and 7 in Table 1, it is seen that although of approximately equal concentrations and specific gravity, the critical velocity of the latter is lower than the former.

Microscopic examination of the first slurry experimented with showed that the particles were unequal in size, were rough and generally larger than in the second sample, which was composed of particles very uniform in size as well as smaller. This also agrees with the second assumption.

It is of interest to note that the value of the friction factor f as computed at the critical velocity was less for Experiment No. 2 of the table, while the viscosity was considerably more than for Nos. 6 and 7. The value of f should vary directly as the density or specific gravity and inversely as the viscosity. It will be seen by comparing No. 2 with Nos. 6 and 7, that the indications are that the variations are in the right direction and roughly proportional.

Conclusions

1 The material can be pumped; the engineering problems involved in pumping are simple.

2 The most economical velocity for pumping is at the critical velocity.

3 The apparent viscosity of the slurry at the critical velocity varied from 24 to 85 times that of water, depending on the amount of solids presented in the slurry.

The author wishes to express his indebtedness to the Louisiana Portland Cement Company, and especially to Messrs. A. D. Stancliff, Superintendent of the New Orleans plant, John L. Porter, Director of the Water Purification Station, for valuable advice and coöperation, and Leon Lassen, Assistant Engineer, for assistance in the experimental work and in the preparation of this paper. Thanks are also due to Prof. J. M. Robert and the Senior Class of the College of Engineering, Tulane University, for help in taking observations.

Analyzing the Indicator Cards of Internal-Combustion Engines

At the Oil and Gas Power Session of the 1926 A.S.M.E. Annual Meeting, last December, a paper was presented by Prof. P. H. Schweitzer, of Pennsylvania State College, dealing with a new method which he had developed for analyzing the indicator diagrams of gas and oil engines. The method consists in drawing a number of tangents to the pressure curve of the card, and a subsequent graphical construction gives the direction of the heat-flow at any point of the expansion or compression line. This tangent method of analysis, according to the author, is simple in execution, is sufficiently accurate, and has been found to be of distinct assistance in testing internal-combustion engines. The paper, which included analyses of some 30 cards taken from a wide variety of engines, was published in the Mid-November, 1926, issue of Mechanical Engineering, pp. 1263–1274. An abstract of the discussion brought out at the meeting follows.

R. Hildebrand[1] wrote that he agreed with the author's method from a scientific standpoint, but questioned its usefulness from a practical standpoint. Only in rare cases could an indicator card be used for the purpose which Professor Schweitzer recommended. It took much experience, a suitable indicator, and a first-class indicator rigging to obtain fairly correct cards. Furthermore, the engine had to be in good working order, i.e., the piston, valves, etc., had to hold tight, and it should have been running for a certain length of time on the load at which the card would be taken, so that the engine would be in a steady condition so far as the heat was concerned. There were so many possible errors which might materially affect the correctness of indicator cards that frequently the analysis would be not only useless but misleading. In such a case it would be better to confine the investigations to determining the B.t.u. used per b.hp-hr., and to measuring the exhaust temperatures. Unless the tangent method of analysis were checked up by the heat consumed per b.hp-hr., it would be of doubtful accuracy.

Robertson Matthews[2] discussed the three methods available for the investigation of internal-combustion-engine efficiency. The use of optical indicators showing flame development and pressure-time relations was instructive in showing variations in the rate of combustion and pressure development, but fell short for numerical evaluations. The analysis of samples of gases taken at successive stages of the working stroke provided definite values on the efficiency of combustion, but had not yet reached successful development. The analysis of indicator cards by some method such as presented by the author appeared at present to be the best method for giving definite numerical information regarding the period of combustion and the magnitude of combustion delay. By his use of a mirror Professor Schweitzer seemed to have largely eliminated the personal element of error in drawing the tangents. To mention possible errors due to the doubtful truth of the indicator card itself, was but to emphasize the great need for further indicator development. Since the accuracy of such analysis depended also upon the accuracy of the determination of the clearance volume, Mr. Matthews had often wished that mathematicians would construct a chart with trilinear dimensions such that the slope of the expansion curve could be determined simultaneously with, but only with, a correct assumption for the percentage of clearance volume. For example, let A and B be the extreme points on a section of the curve to be investigated, let a value for the clearance volume be assumed in per cent of piston displacement, have the absolute pressures and the corresponding per cent piston displacement (instead of piston displacement plus clearance) at A and at B be read off the curve. If now there was a surface or surfaces plotted on which the value of the slope m could be determined when the foregoing values for the pressure and piston displacement at A and B had been taken from the expansion curve, but could be found only when the correct percentage of clearance volume were assumed, then both the need of knowing the clearance volume at the outset and of being misled by an erroneous value having been given for the clearance, would be eliminated. Furthermore, the values for pressures and piston displacements at two points could probably be more easily and accurately determined than a tangent.

[1] Chief Engineer, Diesel Dept., Fulton Iron Works Co., St. Louis, Mo. Mem. A.S.M.E.

[2] Research Engineer, Detroit Edison Company, Detroit, Mich.

Clayton's use of logarithmic paper as a test for the correctness of the percentage of clearance volume involved considerable work. In spite of possible errors in the tangent method of analysis, Mr. Matthews had found deductions from a card of late burning substantiated by test data showing high fuel consumption. Fortunately at that portion of the stroke where late burning was most undesirable, the pressures were such as to give the lowest gas leakages of the expansion stroke.

During compression in an air compressor the curve lay between an adiabatic and an isothermal because of work done upon the air, heat absorbed by cylinder walls, and air leakage. During expansion in an internal-combustion engine there occurred gas leakage, work done by the gases, heat loss to cylinder walls, and possibly some combustion. A case might occur where the effect on pressure drop of the heat lost to cylinder walls and gas leakage was just offset by late combustion, with the result that an apparent adiabatic curve would be discovered. This extreme case showed the difficulties the internal-combustion engine presented for analysis, and it might have been well had Professor Schweitzer presented an indicative value for the fuel consumption of engines likely to have their operation most illuminated by an application of the tangent method.

Certainly any engine showing a fuel consumption of around 0.6 lb. per b.hp-hr. should be legitimate quarry. If a fuel having 15 per cent hydrogen and 82 per cent carbon were used in, say, a large Doxford engine, we should expect early and complete combustion and about 0.37 lb. fuel consumed per b.hp-hr. The hydrogen would burn first, then the carbon. Now it could be shown mathematically that if all the carbon were burned only to carbon monoxide instead of to carbon dioxide, disregarding the small amount of CO_2 due to chemical equilibrium, the heating value of the fuel would be so cut down that the engine would require around 0.66 lb. fuel per b.hp-hr. And there were gas-producer authorities today who declared that carbon burned to carbon monoxide first. Where, then, engines showed fuel consumptions around 0.6 lb., he suggested not only applying the method of tangent analysis but also incorporating a term known as the "gas-producer factor." An engine having 0.66 lb. fuel consumption per b.hp-hr. could, till further refinements were made in the test code to take care of different ratios of compression, be credited with having a gas-producer factor of 100 per cent.

R. W. Angus[3] wrote that, using the same notation as the author had adopted, it was quite easy to show that the heat absorbed by the working fluid during any process represented by pv^n = const. might be found from the equation

$$\delta Q = \frac{k - n}{k - 1} Ap\delta v$$

which equation showed that during expansion, i.e., when δv was positive, the working fluid absorbed heat when the exponent k was greater than the exponent n, and lost heat when k was less than n, and exactly the opposite thing would happen during compression. If, therefore, the expansion or compression line of the indicator diagram were plotted on logarithmic paper, the exponent n for any point on the curve might easily be found by drawing a tangent, and if the corresponding k were known, it was quite easy to see whether heat was being absorbed or rejected.

There was no difficulty about getting this type of information with great accuracy and speed, because the expansion curve might be transferred to logarithmic paper without paying any attention to the pressure and volume scales of the original diagram, and any diagram might be so converted by reading quantities directly with a scale divided into hundredths of an inch. In plotting results the diagram might be made ten times as large just as readily as to the same scale as the original diagram. The main uncertainty in this whole process was with regard to k, which depended on the temperature at the point under consideration, and there was no way of fixing this unless the temperature for some one point on the diagram was assumed. Since very great uncertainty existed as to the temperature of any point in the diagram, there was doubt as to the value of k and hence as to the interchange of heat.

[3] Professor of Mechanical Engineering, University of Toronto, Toronto, Canada. Mem. A.S.M.E.

The author would have done well to say what formulas he had used in plotting Fig. 5, and also what methods he took in getting the temperature diagrams shown at Figs. 6 and 7, because any difference made in assumptions here would make a difference in the position of the k-line. If one were going to study the heat interchanges between the gas and the walls of a cylinder it was always more helpful to know the magnitude of the change as well as its sign, and Professor Angus had found that it was a relatively easy process, if the temperature at one point were assumed, to plot the entropy diagram for an internal-combustion engine, and then all the information that the author had, and a great deal more, was available, since the work diagram had been changed into a heat diagram.

H. Schreck[4] made particular reference to the analysis of the indicator cards of the American-built Diesel engines in Figs. 10, 11, and 12 of the paper. The curves with the exponents as found in these figures did not agree very well with results found by other investigators. For example, the low values of the exponent of expansion in Fig. 10 were attributed to sluggish combustion and overcooled engine, but this was hardly possible with a fuel consumption of 0.448 lb. per hp-hr. Also, the upward rise at c should be considered very carefully. At that point the piston velocity

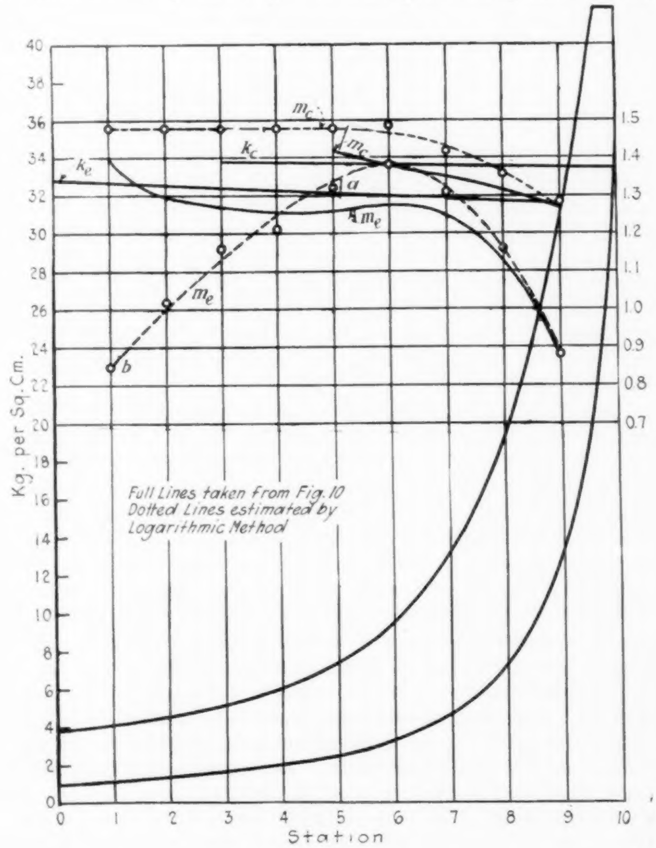


Fig. 1

was low, the cooling surface was large, and only if the fire were coming out of the exhaust pipe would such a rise be possible; but this was not likely at the fuel consumption stated. He had therefore drawn the indicator card Fig. 10 on a larger scale (see Fig. 1), and had analyzed it by the logarithmic method, establishing volume and pressure ratios at various stations.

A polytropic change was expressed by the equations

$$\log \frac{p_2}{p_1} = k \log \frac{v_1}{v_2}; \qquad k = \frac{\log \frac{p_2}{p_1}}{\log \frac{v_1}{v_2}}$$

The values for $\frac{p_2}{p_1}$ and $\frac{v_1}{v_2}$ were plotted on logarithmic paper. Tangents were drawn on the curve at the various stations, and paral-

[4] In charge Design, Combustion Utilities Corp., New York. Mem. A.S.M.E.

lels to these tangents drawn through point 1. For example, taking the tangent at station 7 of the compression curve, the parallel to it intersected the log $\frac{p_2}{p_1}$ line 10 at a distance 5.07. It was then

$$m_e = \frac{\log 10}{\log 5.07} = 1.42$$

The following table gave the values as found in this manner, and also the values as read from Fig. 10 of the author's paper.

	Compression exponent			Expansion exponent		
		m_e			m_e	
Station	Distance	Estimated	Reading, Fig. 10	Distance	Estimated	Reading, Fig. 10
1	4.75	1.48	.....	15	0.85	1.4
2	4.75	1.48	.....	9.55	1.02	1.3
3	4.75	1.48	.....	7.28	1.16	1.27
4	4.75	1.48	.....	6.68	1.21	1.26
5	4.75	1.48	1.42	5.75	1.32	1.26
6	4.68	1.49	1.38	5.29	1.38	1.275
7	5.07	1.42	1.35	5.82	1.31	1.245
8	5.45	1.36	1.315	7.23	1.16	1.13
9	6	1.283	1.265	14	0.874	0.875

These results were entered in Fig. 1 and showed very little agreement with the values found by the tangent method. They had also been checked by the well-known formula

$$n = \frac{\log p_1 - \log p_2}{\log v_2 - \log v_1}$$

and the agreement with the values of the logarithmic method was very close. Mr. Schreck did not claim that his curves were absolutely correct, because a slight difference in his measuring an indicator card from a small print was very unsatisfactory. He suggested, therefore, that the author, as a supplement to his paper, either supply the correct readings of pressures, volumes, and compression ratios, of all the indicator cards, or that he add to his paper, as a check, the logarithmic analysis of the indicator cards of the American-built Diesel engines, because they really showed up worse than the cards of any other make, and therefore made a further investigation desirable.

The author, Professor Schweitzer, in his closure said that he had anticipated that most of the discussers of the paper would question the accuracy of the tangent method. However, all of the questions raised belonging to this category had been investigated and answered in Appendices B, C, and D of Bulletin No. 35, of the Enginering Experiment Station of The Pennsylvania State College, where many pages were devoted to the effect of various errors. Among others, errors caused by the inaccurate drawing of tangents, by a wrong estimation of the clearance volume, by the imperfectness of the gas, by cylinder leakage and various imperfectnesses of the indicator mechanism were investigated and their effect on the graphical indicator analysis numerically evaluated. The effect of all of the errors investigated had proved to be small enough not to affect seriously the accuracy of the analysis, provided the indicator was not unreasonably inaccurate. Regarding indicator errors, he found that cord stretching was perhaps the most serious; this affected the results by as much as 26 per cent in extreme cases. Other indicator errors, such as pencil friction, piston friction, linkage friction, etc., caused only insignificant final errors in the analysis. He agreed, however, with Mr. Hildebrand that a check by the heat-consumption figure was always desirable.

Mr. Matthews had asked how the clearance volume could be determined in various cases. In the bulletin referred to it was shown that a slight error in estimating the clearance volume did not affect the accuracy of the analysis appreciably. For instance, in a Diesel engine, a 10 per cent error in the clearance volume would affect the m's by less than 0.5 per cent. However, the author had published an improved method for approximating the clearance volume of internal-combustion and steam engine in *Engineering*, July 18, 1924.

The author did not agree with Professor Angus in his criticism of the formula he had used. Neither did he believe that the formula Professor Angus had quoted was correct unless n were constant during the change, which was seldom the case. Appendix A of the bulletin referred to dealt with this question.

Regarding Professor Angus' and Mr. Schreck's suggestion that the logarithmic method proposed by Clayton gave more reliable results, the author said that, comparing it with the tangent method, he did not see any advantage in the former. The logarithmic method also involved the drawing of tangents, but it had the disadvantage of necessitating a scaling, often inaccurate, of the diagram which was not necessary in the tangent method. Actual experiments showed that tangents could be drawn to a general curve by an inexperienced person with an accuracy of less than one degree, and the mathematical analysis had shown that one degree would not make a larger error in the m than 5 per cent except near the dead center, where the indicator analysis could not be applied because of open valves. In several cases when the indicator diagram was large enough to minimize the inaccuracies of scaling or the pressure values were given numerically (see pp. 57–61, Bulletin No. 35), comparisons had been made between the logarithmic and tangent methods and good agreement obtained. The disagreements Mr. Schreck had noted were partly due to the inaccuracy in scaling the pressure values from the diagram, and partly to the fact that the logarithmic method tended to smooth out and mask the variations in the m's while the tangent method showed them on a large scale.

The plotting of entropy diagrams from the indicator diagrams as a substitute for the indicator analysis seemed to be too complicated to the practical engineer and even to the author himself with the gas charts available. To the other question of Professor Angus regarding the formulas used in plotting Fig. 5, the author referred him to p. 24 of the bulletin mentioned.

Regarding Fig. 10, to which Mr. Schreck had made reference, the author pointed out that Mr. Schreck must be mistaken in stating that he (the author) attributed the values of m to sluggish combustion and an overcooled engine. On the contrary, he had spoken of a quick ignition and uniform cooling. The upward rise at c was quite natural.

Comparison of the results of the analysis with actual performance in fuel consumption had invariably confirmed the accuracy of the method. In Appendix E of Bulletin No. 35 a comparison had been made between the results obtained by the tangent method and those of a thermodynamic analysis made by Rosecrans and Felbeck at the University of Illinois and published in its Bulletin No. 150. This comparison, including speed of combustion, afterburning, cooling, etc., had confirmed more than anything the reliability of the tangent method.

The Institute of Chemistry

THE First Institute of Chemistry is to be held during July, 1927, at Pennsylvania State College. This has been arranged by the American Chemical Society for the purpose of offering a series of lectures and demonstrations whereby those in attendance may be brought quickly up to date in fields both within and outside their own specialty, and to afford facilities for teachers to acquire the lastest information in chemical science as well as to benefit from the contacts with the industrial and consulting professional chemists.

It is planned that teachers and others desiring to do so can take the stated courses in chemistry throughout the summer school and receive credit therefor, so that in a combination of the Institute of Chemistry of the American Chemical Society and the regular summer-school courses of Pennsylvania State College the requirements will be met.

The Chemical Foundation, Inc., and the Pennsylvania State College have agreed to furnish the funds for the first session. Northwestern University has requested the privilege of being the second university to coöperate with the society, and the session of 1928 is to be held at Evanston.

A Correction

IN THE Mid-May issue of MECHANICAL ENGINEERING, the caption at the bottom of page 550 should read: "Trade-School Certificates Awarded on a Time Basis;" that on page 551, "Trade School Apprenticeship Certificates Awarded on a Proficiency Basis;" and that on page 556, "Coöperative Apprenticeship Certificates."

Stresses Occurring in the Walls of an Elliptical Tank Subjected to Low Internal Pressures

By WM. M. FRAME,[1] SHARON, PA.

The paper describes a test made on an elliptical tank, and presents an analysis, based on experimental data, from which the stresses set up in the walls may be calculated. To account for the fact that the tank appeared stronger than was to be expected from usual assumptions, it was assumed that the wall acted as a beam supported at the top and bottom of the tank. The analysis from this point of view yields results which check with reasonable accuracy the experimental data.

THE Westinghouse Electric and Manufacturing Company uses elliptical tanks to enclose some types of transformers. The walls of these tanks must withstand the hydrostatic oil pressure due to the tanks, being filled with oil, and an additional air pressure of 10 lb. per sq. in. These tanks, designed empirically at first, were used successfully, yet approximate calculations indicated stresses high enough to cause failure of the walls. It seemed necessary to obtain some experimental data in order to arrive at the correct theoretical assumptions, and accordingly, a tank was taken from stock and subjected to a test.

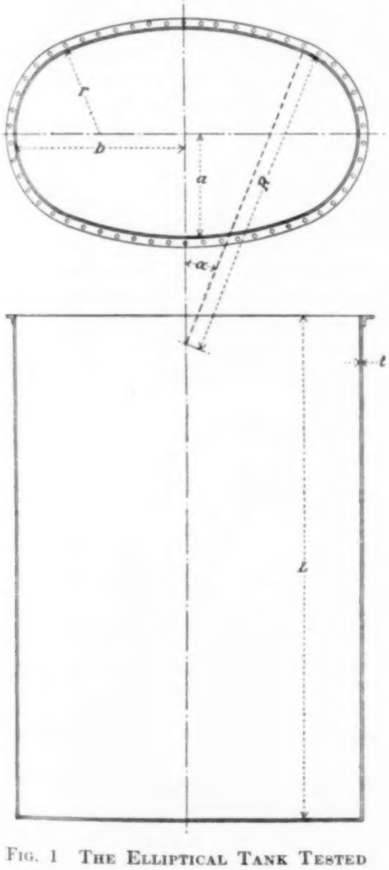


Fig. 1 The Elliptical Tank Tested

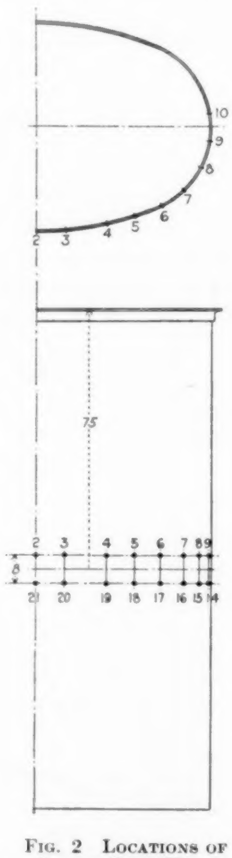


Fig. 2 Locations of Strain-Gage Holes

The Tank Tested

A general layout of the tank is shown in Fig. 1. The walls are made of boiler plate, all seams being arc-welded. The dimensions of the tank are as follows:

L = height = 144 in.
a = semi-minor axis = 30 in.
b = semi-major axis = 50 in.
t = thickness of wall = $^3/_8$ in.

The ellipse was constructed with two radii, bearing a certain relation to the major and minor axes.

$$a = 0.6b \qquad R = 3a$$
$$\alpha = 22 \text{ deg. } 37 \text{ min.} \qquad r = 5a/6$$

A braced cover was constructed and this was bolted to the tank flange. A cemented cork gasket was placed between the cover and the flange. An air line, with a valve and pressure gage, was fitted to the cover.

The Tests

The purpose of the test was to measure the stresses and deflections set up in the walls of the tank due to hydrostatic oil pressure and additional air pressure. The stresses were measured by means of a Berry strain gage placed in holes drilled in the wall, located as shown in Fig. 2. These holes were $^1/_{32}$ in. in diameter and $^1/_{16}$ in. deep, placed 8 in. apart. To measure the deflections, wooden standards were erected close to the tank: one on the major axis, one on the minor axis, and one 32 in. from the major axis measured along the periphery.

Extensometer readings and deflection readings in both the vertical and horizontal directions were taken with the tank empty and at zero gage pressure; then with the tank filled with oil weighing 0.0303 lb. per cu. in., and zero gage pressure; and then at gage pressures of 0, 1, 3, 5, 7.5, 10, 13, and 14.5 lb. per sq. in. It was found that the extensometer readings taken in the horizontal direction between holes 21 and 20, Fig. 2, were small compared with the vertical elongation between holes 2 and 21. Also, the elongation in the horizontal direction was fairly constant all around the periphery, as might be expected if the cross-section of the tank had been circular instead of elliptical. In contrast with this, the strain of the metal in the vertical direction varied from maximum tension between holes 2 and 21 to maximum compression between holes 9 and 14. The extensometer readings for the corrected gage pressure of 16 lb. per sq. in. are given in the second column of Table 1.

TABLE 1

Position of extensometer (see Fig. 2)	Extensometer reading corrected to zero	Extensometer reading, in.	Strain of metal, in. per in.	Fiber stress, lb. per sq. in.
2–21	0.016 tension	0.0030	0.000375	12380
3–20	0.0125 tension	0.00234	0.000292	9630
4–19	0.006 tension	0.00113	0.000141	4650
5–18	rejected	....	....	.....
6–17	0.007 comp.	0.00131	0.000164	5410
7–16	0.0135 comp.	0.00253	0.000316	10420
8–15	0.0135 comp.	0.00253	0.000316	10420
9–14	0.0140 comp.	0.00264	0.000327	10800

The stresses obtained from the extensometer readings were calculated in the following manner: Consider an element of the tank wall and assume that it is loaded in both directions, letting

p_y = unit stress in the horizontal direction around the tank
p_z = unit stress in the vertical direction
e_y = unit strain in the horizontal direction
e_z = unit strain in the vertical direction
δ = Poisson's ratio = 0.3 for steel, and
E = modulus of elasticity = 30×10^6 for steel.

Then, according to Poisson's theory:

$$e_z = \frac{p_z}{E} - \frac{\delta p_y}{E}$$

$$e_y = \frac{p_y}{E} - \frac{\delta p_z}{E}$$

Adding these two equations and rearranging:

$$p_z = \frac{E}{1 - \delta^2}(e_z + \delta e_y)$$

[1] Westinghouse Electric & Manufacturing Company. Jun. A.S.M.E.
Presented at the Annual Meeting, New York, December 6 to 9, 1926, of The American Society of Mechanical Engineers. Slightly abridged.

As the value of e_y was small compared with e_x, it can be neglected, and the equation becomes

$$p_x = \frac{Ee_x}{1 - \delta^2} \qquad [1]$$

The stresses in Table 1 were calculated, using Equation [1].

THE BEAM THEORY

The fact that the stresses decreased from maximum tension at the minor axis down to zero, and then increased to maximum compression at the major axis, seemed to indicate that the wall acted as a beam supported at the top and bottom of the tank. To

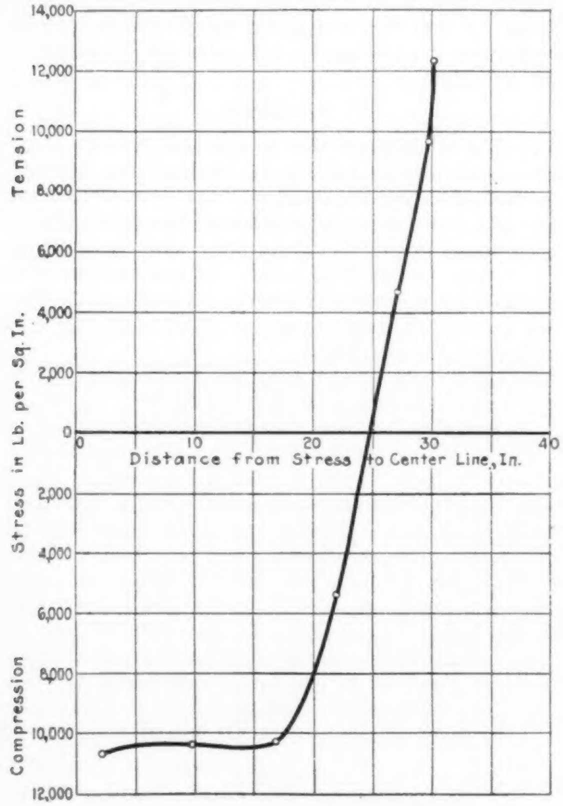


FIG. 3 LOCATION OF NEUTRAL AXIS

get a clearer picture of this, the stresses were plotted against the distances from the major axis to the points at which the stresses occurred. The resulting curve is shown in Fig. 3. Part of the curve is fairly straight, and indicates that at least a portion of the wall did act as a beam supported at the top and bottom of the tank. The curve also indicates that the neutral axis existed at a distance of 24.8 in. from the major axis.

The next logical step was to determine how much of the wall acted as a beam, assuming that the neutral axis existed at a distance of 24.8 in. from the major axis. In order to accomplish this, the following equation was derived. Referring to Fig. 4, it was assumed that the part of the wall acting as a beam ends at A, as indicated by the angle θ. Also the distance from the center line to the neutral axis is d. Then

$$d = \frac{(r^2 + rt)\sin\theta + \sin\alpha[(R - r^2) + t(R - r)] - \alpha\left(R + \frac{t}{2}\right)(R - a)}{\left(r + \frac{t}{2}\right)\theta + \alpha(R - r)} \qquad [2]$$

Using the value of d already obtained from the curve in Fig. 3, θ was calculated by means of Equation [2], and found to be 61 deg. In the equation, θ and α are in radians.

The above procedure was repeated for the stresses occurring at the various pressures with which the tank was tested. That is, the stresses were plotted against the distances from the major axis, the distance from the major axis to the neutral axis was thus ascertained, and the angle θ was calculated from Equation [2]. Although the value of θ thus obtained varied from 49 deg. to $66^1/_2$ deg., it was found that the average value was about 61 deg. This demonstrated rather conclusively that a certain part of the wall acted as a beam supported at the top and bottom, irrespective of the pressure applied. The next logical step was to assume such a beam, and, knowing the loading, to calculate the stresses set up. If these calculated stresses checked the test stresses, the assumptions would be proved correct, and the analysis as a whole would be acceptable for future use.

In order to obtain the moment of inertia of the beam, Equation [3] was derived.

$$I = t\Bigg[R^2\left(R + \frac{3t}{2}\right)(\sin\alpha\cos\alpha + \alpha) - 4R(R + t)(R - a + d)\sin\alpha + 2\alpha\left(R + \frac{t}{2}\right)(R - a + d)^2 + r^2\left(r + \frac{3t}{2}\right)\{\sin\theta\cos\theta + \theta - (\sin\alpha\cos\alpha + \alpha)\} - 4dr(r + t)(\sin\theta - \sin\alpha) + 2d^2\left(r + \frac{t}{2}\right)(\theta - \alpha)\Bigg] \qquad [3]$$

where I = moment of inertia of that part of wall acting as a beam
θ = angle as shown in Fig. 4 = 61 deg. in this case
α = angle of construction as indicated in Fig. 4
r = small radius
R = large radius
t = thickness of the wall
a = semi-minor axis
d = result obtained from Fig. 3.

All dimensions are inside measurements in inches and all angles are in radians. Solving Equation [3], using the dimensions for this tank, the value of I was found to be 1010 in.^4

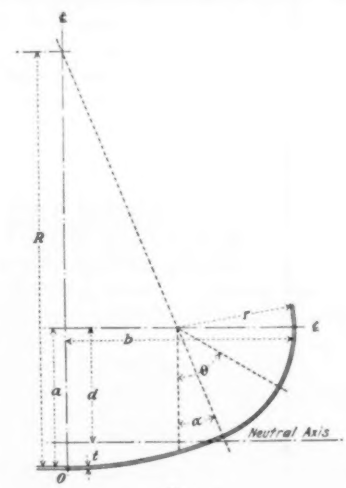


FIG. 4

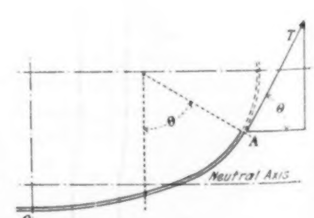


FIG. 5

The loading of the beam was determined by multiplying the horizontal width of that part of the wall considered as a beam by the average acting pressure, and subtracting from this the tangential pull of the remaining part of the wall. Referring to Fig. 4, let the horizontal width of the wall from O to A equal H.

$$\begin{aligned} H &= 2[R\sin\alpha + r\sin\theta - r\sin\alpha] \\ &= 2\sin\alpha(R - r) + 2r\sin\theta \end{aligned}$$

Let P = air pressure in lb. per sq. in.
K = weight of oil in lb. per cu. in.
L = total height of the wall.

The tank is to be considered full of oil. Then the total pressure due to the air and the oil at the middle of the beam will be

$$\text{Total pressure} = P + \frac{Kl}{2}$$

Let V = the apparent load per unit length of beam. Then

$$V = \left(P + \frac{Kl}{2}\right)[2 \sin \alpha(R - r) + 2r \sin \theta] \quad [4]$$

The air pressure is a uniform load over the length of the beam, but the oil pressure is in the form of a triangular load. However, for the purpose of calculating the moment at the center of the beam, the value of V, as obtained by Equation [4], may be used with less than one per cent error. The error increases somewhat for the smaller values of P.

There is another force acting on the assumed beam which must be subtracted from the apparent load. This force is the tangential pull from that part of the wall not considered acting as a beam. Referring to Fig. 5, it can be seen that the edge of the beam, as indicated by point A, lies well within that part of the wall formed by the radius r. It was natural to conclude that this part of the wall would be stressed as a circular ring of radius r subjected to an internal pressure of $P + \frac{Kl}{2}$ lb. per sq. in. Using this assumption, the tangential load T was calculated.

$$T = \left(P + \frac{Kl}{2}\right)r \quad [5]$$

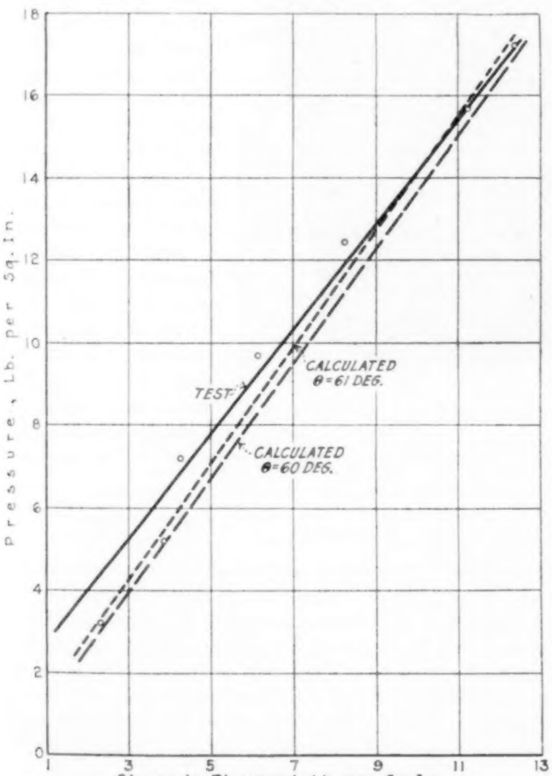


FIG. 6 TEST VALUES AND CALCULATED VALUES OF STRESSES AT DIFFERENT PRESSURES

T = tangential load per unit length of beam. The vertical component of T is the effective force on the beam, which must be subtracted from the apparent load V to give the resultant load on the beam. Let Q = the resultant load per unit length of beam. Then

$$Q = V - 2T \sin \theta \quad [6]$$

Assuming that the beam is supported at the top and bottom of the tank, the moment M at the center is

$$M = \frac{QL^2}{8} \quad [7]$$

$$S = \frac{Mc}{I} \quad [8]$$

in which c = the distance from the neutral axis to the extreme fibers.

Using the value obtained for I and solving Equations [4], [5], [6], [7], and [8], the calculated value of the stress S was determined. The stress was calculated for the extreme fibers in tension, using a net pressure of 15 lb. per sq. in. The results are as follows:

I = 1010 in.^{4}
V = 1620 lb. per sq. in.
T = 432 lb. per in.
Q = 865 lb. per in.
M = 2,245,000 lb-in.
C = 5.5 in.
S = 12,200 lb. per sq. in. (tension).

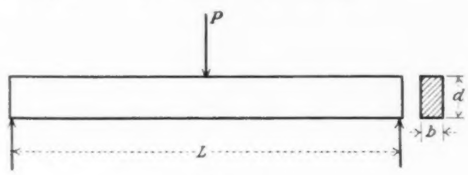


FIG. 7

This calculated value of stress should correspond with the test stress obtained at position 2–21 of Fig. 2. Referring to Table 1, it is seen that the value of the test stress was 12,380 lb. per sq. in. The calculated stress checked the test stress so closely as to be well within the limits of accuracy.

The stresses were calculated for other values of pressures and also assuming that β equaled 60 deg. The resulting curves are shown in Fig. 6.

Using these same values of load, etc., it should be possible to check the test deflection obtained at the minor axis.

$$f = \frac{5QL^4}{384EI}$$

f = 0.160 in.
f = deflection at the center of the beam.

The actual test deflection at this point was 0.28 in. These values are not in good agreement; however, it must be remembered that the observed deflections were quite inaccurate.

An explanation can be given to account for the fact that the extreme fibers in compression were not stressed as much as the beam theory indicated. The extreme fibers in compression were farther from the neutral axis than the extreme fibers in tension, and the observed stress should have been greater. However, an examination of Table 1 will reveal that the maximum compressive stress was less than the maximum tensile stress. It seems probable that the wall buckled under compression, thus relieving the stress.

CHECK OF THE BEAM THEORY

It is interesting to note the effect a certain percentage increase or reduction in all linear dimensions will have on the stresses in the walls, assuming that the pressure is kept constant. First, for simplicity, consider a beam of rectangular cross-section, Fig. 7, supported at the ends, with a concentrated load at the center.

$$M = \frac{Pl}{4}; \quad S = \frac{M}{Z} = \frac{3P}{2}\frac{L}{bd^2}$$

Assume that all linear dimensions are doubled.

$$S_1 = \frac{3P}{2}\frac{2L}{2b4d^2} = \frac{3P}{2}\frac{L}{4bd^2}$$

It is seen that the stress has been reduced to one-fourth of the original value.

Now consider the tank and assume that all linear dimensions are doubled.

$$S = \frac{QL^2}{8}\frac{C}{I}$$

As can be seen from Equations [4], [5], and [6], the load Q depends on the first power of a linear dimension. From Equation [2], the

distance to the extreme fibers C also depends on the first power of the linear dimension; every term in the numerator contains a linear dimension squared, and every term in the denominator contains a linear dimension. Every term in Equation [3] for the moment of inertia I contains a linear dimension of the fourth power. So, if we hold the air pressure constant and double all the linear dimensions, the stress can be obtained from the following equation.

$$S = \frac{2Q4L^2 \times 2C}{8 \times 16I}$$

But this stress is the same as that which occurred in the original tank. So it is seen that the same stresses will be obtained in a tank model, built with all dimensions proportional to the original tank, by using the same air pressure.

In order to check the theory in the preceding paragraph, a tank model was built and tested. All dimensions were reduced from

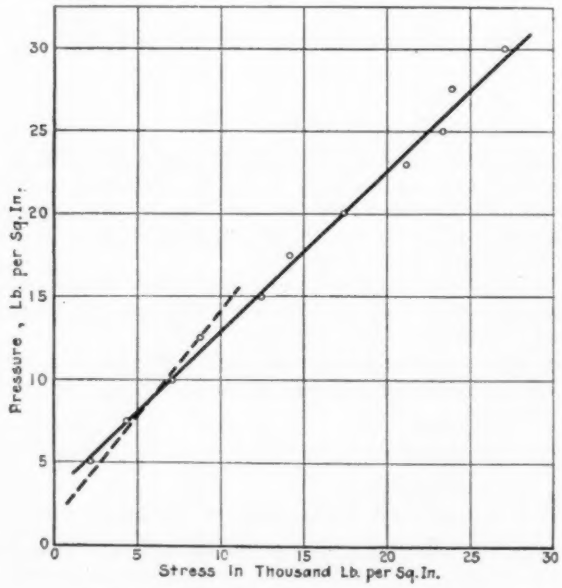


Fig. 8 Stress in Walls of Tank Model

those of the original tank in the ratio of 1 to 0.136. The walls were made of steel sheet, 0.051 in. thick. The top and bottom were made of $^1/_4$-in. steel plate to eliminate bracing. The walls were heavily gas-welded to the top and bottom.

Air pressure was applied to this tank model by means of a tire pump. Extensometer readings were taken in prick-punch marks located at the center of the wall, as indicated by position 2–21 of Fig. 2. Readings were obtained up to 30 lb. per sq. in. air pressure. The results are shown by the full-line curve in Fig. 8. The dotted curve is a reproduction of the curve in Fig. 6, and indicates the stresses obtained in the walls of the large tank. The difference between these two curves, which are in good agreement, can be accounted for by the personal error in reading the strain gage.

It was believed that some interesting results would be obtained by testing this tank model to destruction. Accordingly, the tank was filled with oil and pressure was applied by means of an oil pump. Bulging of the walls could be detected by close observation at a gage pressure of 30 lb. per sq. in., and this bulging increased slowly until, at 60 lb. per sq. in., the wall buckled in two places, as shown in Fig. 9. A pin hole opened up in the vertical wall seam, and oil was sprayed out in a fine mist. The pressure was increased and, at 80 lb., the seam opened up, relieving the pressure. The beam action is clearly shown. The fibers in compression failed, due to buckling, and the fibers in tension were permanently deformed, as shown in Fig. 9.

Discussion

S. W. Miller[1] wrote that in some cases the design of the tank might be such that the beam would have to be considered as being at least partially fixed at the ends, instead of being only supported at those points, which would introduce an additional longitudinal tensile stress in the beam. Mr. Frame had only considered the strain in two directions. In reality there was a strain in all three dimensions, and while that in the third dimension might not be of much importance in thin-walled tanks, yet it had a material effect on the stresses in the cases where the shell was of considerable thickness compared with the diameter. The kind of work done by Mr. Frame indicated the increasing interest than was being taken in more careful design, and showed the necessity of using the strain gage much more liberally than had been done in the past. With regard to the method of test, it was much safer to use hydrostatic pressure for the preliminary test, that was, the one to determine the strength of the tank. After this had been found to be satisfactory, air pressure could be used if necessary for the location of small pin holes or other very minor defects. There had been a number of accidents from the use of air for testing to destruction.

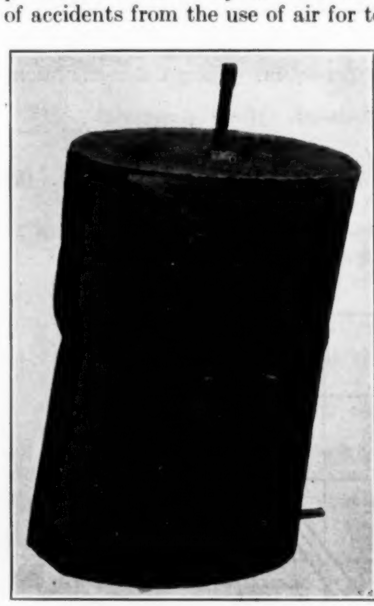


Fig. 9 The Tank Tested to Destruction

S. Timoshenko[2] stated that the theory presented by the author had proved very useful. A. L. Kimball, Jr.,[3] thought that a qualitative consideration of extreme cases was useful in cases of this sort. For example if the tank were circular there would be no compression anywhere along the circumference. The stress due to the pressure would be all longitudinal.

Joseph G. Bercsi[4] wrote that the calculated stresses given by the author in Table 1 consisted of two components if the beam theory were accepted: (1) a stress which was caused by the pressure, parallel with the vertical axis of the tank, and which might be calculated by dividing the whole pressure in this direction by the cross-section of the tank wall, i.e.,

$$p_1 = \frac{abp}{\text{cross-sect.}} = 725 \text{ lb. per sq. in.}$$

figuring the area of the bottom as that of an ellipse having the semi-axes a and b; and (2) the stress which derived from the bending. For this reason if it was desired to obtain the tension caused by the bending only, it was necessary to subtract the tension p_1 from that in the end of the minor axis, because both were tensile stresses and had the same direction. This meant subtracting from the fiber stresses given in Table 1 the value $p_1 = 725$, and noting that the compression stresses had a negative value. Therefore the bending stress at the end of the minor axis was really $S = 12{,}380 - 725 = 11{,}655$ lb. per sq. in. This was the value that had to be compared with the bending stress $S = 12{,}200$, figured by means of the beam theory. According to this, the neutral axis had to move further away from the major axis. But these changes affected the results only to a small extent.

Having the neutral axis and the cross-section of the tank below the neutral axis, the author had calculated the cross-section above the axis and obtained the angle $\theta = 61$ deg.

If one figured the compression stress at the point A (see Fig. 5 of the paper) by the beam theory, he would get $S = 28{,}200$ lb. per sq. in., instead of 10,400 lb. This difference was very considerable, and it could not be explained by the assumption that the wall buckled under compression, thus relieving the stresses.

[1] Union Carbide and Carbon Research Laboratories, Long Island City, N. Y. Mem. A.S.M.E.

[2] Engineer, Research Department, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. Mem. A.S.M.E.

[3] Research Engineer, General Electric Company, Schenectady, N. Y. Assoc. A.S.M.E.

[4] Westinghouse Electric & Manufacturing Co., Sharon Works, Sharon, Pa. Assoc-Mem. A.S.M.E.

Having all the necessary data, the author wanted to prove his theory by calculation. To be able to use the beam theory, he had made some assumptions which, however, could not be sustained.

First of all, the assumption that the wall was supported at the bottom and the top seemed to be incorrect. The tank wall was clamped at the top and bottom, because the tangents at these points had to be always parallel with the vertical axis of the tank. Therefore at these points there was not only the reaction stress but a bending moment also which had to be taken into consideration when calculating.

Besides, the author took into account only the vertical components of the internal pressure p and the bending moments of these pressures around the assumed supports. The tangential component T was calculated by the author in the same way. To figure correctly with the tank wall as if it were supported at the ends, it was necessary to calculate as for a supported curved plate, because its length, 144 in., and width, somewhat less than 100 in., did not differ too much. In this case all the pressures p were to be resolved into two components, p_x and p_y (see Fig. 10). If two p_y stresses were applied at the end of the minor axis, a couple would result which would have a tendency to bend the plate around the axis B. The remaining p_y would have a tendency to bend the tank wall around the supports. The same was the case with the components p_x, which had a bending moment around the axis B. The remaining p_x would cause tension only at the section B because of the symmetry of the tank. Both components of the tangential stress T also had to be considered.

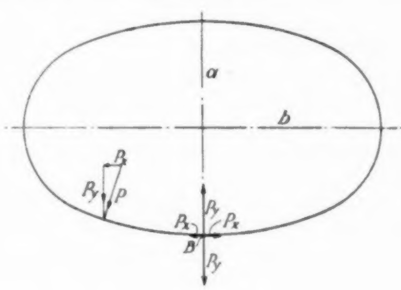


FIG. 10

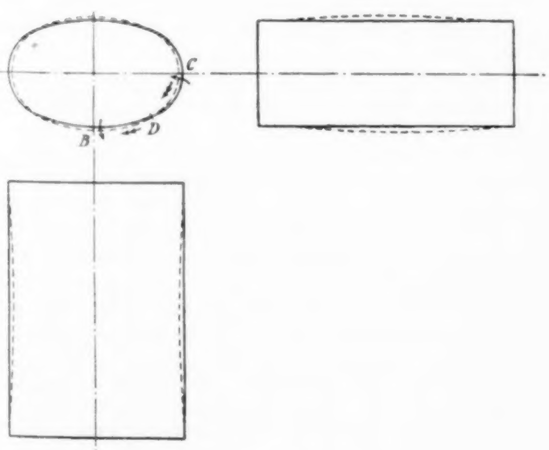


FIG. 11

The tangential stress T was assumed by the author to be equal to $\left(P + \dfrac{Kl}{2}\right)r$. This was not correct. In the case of an elliptical tube the tangential stress would be equal to pb at the end of the major axis and to pa at the end of the minor axis where p was the whole internal pressure. At the point A (see Fig. 5) this value would be $p\sqrt{x^2 + y^2}$, if x and y were the coördinates of point A. In the present case these values were smaller, because the height of the tank was only a little larger than the major axis.

Another assumption, which was not correct, was where the author substituted the part of the tank wall not considered acting as a beam with one stress only without considering the bending moment. In case of an elliptical tank this could not be done, because there was always a tangential stress and a bending moment along the perimeter (except on four points, as shown later).

The hypothesis that the perimeter expanded all around equally could not be explained by the beam theory, because if the tank wall were bent according to the beam theory, the perimeter could not increase but should decrease, this part of the tank wall being perpendicular to the direction of the tension.

Considering these facts, we could not expect that the compression stress for the point A calculated from the test and the approximately measured deflection at the end of the minor axis would be in agreement with the figured values. The beam theory was used that way, and gave in this particular case a good approximate value for the tension at the end of the minor axis, but the results

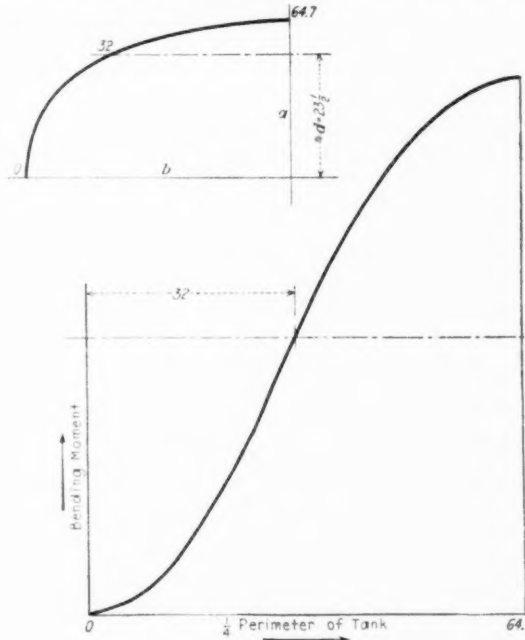


FIG. 12

for other points and the deflection did not correspond with the test. This showed, that the beam theory was not correct.

Mr. Bercsi also discussed the deformation of the tank wall from another viewpoint. First of all he considered the action of the internal pressure on the tank wall. It was clear, that the elliptical tanks, when subjected to internal pressure, tended to become round. This was shown on Fig. 11. It was obvious that the minor axis tended to increase and the major axis to decrease. We would imagine that there were four lines around the tank wall (generatrix) parallel with its vertical axis (denoted by D in Fig. 11) around which the tank wall was turned as the single-headed arrows showed. For this reason the lines on the tank wall which were parallel with the vertical axis and to the left of the point D had to increase in length during the deformation, and those to the right had to decrease. The points B and C had to move always on the minor and major axes because of the symmetrical shape of the ellipse. From this reasoning it might be concluded that the line BD had to elongate and the line BC to decrease if the point D did not move during the deformation. But Mr. Frame's test had shown that the point D would move nearer to B under the action of traction and compression (shown by the double-headed arrows), so that in this direction there was a slight elongation only all around the periphery.

Explaining the deformation this way, it was evident that the lines on the periphery at the end of the minor axis had to increase most and those at the end of the major axis to decrease, taking into consideration always the vertical lines. Between these lines there had to be one which did not change its length, as the test had shown (see point D, Fig. 11). Accepting this deformation, the tank wall could not buckle at the point C or near there, because there was no active compression stress but a bending moment which caused the deformation. With this explanation we should not get a com-

pression stress of 28,000 lb. per sq. in. for the point A, as figured by the beam theory.

With the above assumptions, which were correct for elliptical tubes, the writer figured the location of the point D as the one where no bending moment existed. Using the Westphal equation or the graphical method described in Fuller and Johnston's Applied Mechanics, pages 466 to 470, he obtained 23.5 for distance d instead of 24.8, as the author did (see Fig. 12). But this result was in agreement with the author's test if we considered that the measurements were taken about 8 in. apart in the horizontal direction, and the point where deformation was not measured was 5–18. The writer supposed that if these points were 3 in. to the right from their present place, no deformation could be measured there too. And this was about the place which coincided with the figured point D. From this it might be seen that the lines around which the tank wall was turning during the deformation were at the same place as in the case of a long tube. It was just an accident that this point D was approximately the common point of the large and small circle, that is, it was about 22 deg. from the minor axis.

The deformation of the elliptical tank could readily be explained in this way, but the equations for the elliptical tubes could not be used in this case, because the height of the tank was small in comparison with the width.

The Smithsonian Institution

By FRANK TAYLOR,[1] WASHINGTON, D. C.

IT IS SAFE to say that every professional person in America is familiar with some phase of the work of the Smithsonian Institution. Even more safely can it be said that only a meager few have more than the vaguest idea of the Institution's origin, scope, or official status.

The story of the origin of the Institution is the story of the life of the founder, the Englishman James Smithson. Born the illegitimate son of the Duke of Northumberland, Smithson overcame a natural bitterness toward life and devoted himself to a career of service. A successful chemist and mineralogist, he was admitted to the Royal Society at the age of twenty-one. Through his whole life he sought to promote general education and to make scientific knowledge popular and available. On the subject of education he expressed himself thus: "No ignorance is probably without loss to man, no error without evil." This ruling motive led him to plan the establishment of a national educational institution at London, to be under the direction of the Royal Society. Later, disgusted by the Society's refusal to publish some of his work, he changed his plans and apparently gave up the scheme. To his belief in the soundness of the idea and his wish to see it realized, James Smithson owes his lasting fame, for to the United States of America he left the English equivalent of about $550,000 to found at Washington, under the name of the Smithsonian Institution, an establishment "for the increase and diffusion of knowledge among men."

Such a gift was without precedent and required some consideration prior to acceptance. There were many opposed to the gift. Some believed it beneath the dignity of the country to accept a gift from an individual, and others thought that James Smithson sought immortality at too cheap a price. Congress was finally induced to accept, after which a period of eight years was required to design the institution, suggestions for which ranged from an astronomical observatory to an agricultural experimental station. The charter as finally drawn recognized that the money could never become a part of the revenues of the United States but had been accepted only in trust for a specific purpose, and described an establishment responsible for the duty of carrying out this purpose.

To the seemingly impossible task of usefully administering such a trust, the first Board of Regents elected Joseph Henry the first secretary of the Smithsonian Institution. Professor of physics at Princeton, a pioneer in the field of electricity, whose discoveries made practicable the electric telegraph, and independent discoverer of laws of magnetomotive force, contemporary with Faraday, Joseph Henry was unanimously recognized as the foremost American scientist. That a man of such achievement should have lacked the administrative and organizing ability would have been excusable, but here was a man so able that he could not fail. The policy and organization developed by Henry, together with his personal sacrifice of career to direct the program, entitle him to an equal share with Smithson in the success and glory of the Institution. As knowledge is increased by research, Joseph Henry considered it the duty of the Institution to encourage and subsidize investigations, to promote original research, and to encourage investigators by publication of their results. The diffusion of knowledge was to be accomplished by free publication and distribution of popular accounts of new discoveries, and by international exchange of scientific literature. So sound is this program that for eighty years the Institution has followed it to success and international leadership in science. Research in every field has been promoted, and scientists the world over have received encouragement and financial assistance from the Institution. Nine activities originated by the Smithsonian have resulted in the formation of independent Government bureaus, seven of which have remained under the direction of the Institution. Among the nine are the Weather Bureau, the Bureau of Fisheries, the Astrophysical Observatory, and the United States National Museum. Among the many historic individual achievements subsidized by the Institution are Morley's determination of atomic weights, Schumann's work on the ultra-violet ray, and Langley's contribution to the science of flying. In conjunction with these activities has been the program of publications. The bureaus and the National Museum receive appropriations from Congress, part of which pay for the publication of the results of their work. The Smithsonian series, though, are financed entirely from the private funds of the Institution and are distributed free of charge to more than 1500 libraries throughout the world. In these series is published new knowledge gained by members of the staff and outside workers. Annually, the report of the Institution contains reprints of material which best shows the advance made in science during the year. These articles impartially cover the entire field and are of a generally comprehensible style. The publications most familiar to engineers are the Smithsonian Mathematical and Physical Tables. These tables are being constantly revised and extended, and new editions appear as often as funds permit. Closely related to the program of publication are the International Exchange and the Catalogue of Scientific Literature. These are now Government bureaus which the Institution directs. The function of the first is that of exchanging for the publications of subscribing foreign governments, universities, and scientific societies, the publications of our Government and associated institutions. The literature received here under this plan is transferred to the Library of Congress where it forms part of the Smithsonian Deposit, a scientific library of over 700,000 volumes. The Bureau of the Scientific Catalogue is the United States branch of an international organization engaged in the compilation and publication of a perpetual index of scientific literature.

Any sketch of the Smithsonian Institution must fail to completely treat the subject, and no attempt has been made to show more than a typical cross-section of the activities. The major aim has been to establish the singular uniqueness of the "Establishment." Working on a scanty private endowment, directed by the highest officials of the Government, and, in turn, directing seven governmental bureaus, the Institution has practically no official status. On the other hand, no organization has so good a claim to the unofficial position of Leader in Science. Certainly the Institution is fulfilling its trust, increasing and diffusing knowledge to a very high degree.

[1] Aid, United States National Museum. Member, George Washington University Student Branch of the A.S.M.E.

The Design of Dished and Flanged Pressure-Vessel Heads

By A. B. KINZEL,[1] LONG ISLAND CITY, N. Y.

THE formulas generally used for this purpose are modifications of the basic formula for the strength of a hollow sphere subjected to internal pressure. The formula in the power-boiler section of the A.S.M.E. Boiler Code has been in that Code since 1914 and the results of its application seem to have been generally safe, though there have been some head failures.

This formula was devised at a time when boilers were customarily made of thinner plate than at present, and it was probably intended for use in connection with drums rather than for boiler shells. The addition of $^1/_8$ in. to the formula thickness to take care of a manhole in the head under these conditions was probably well enough, as $^1/_8$ in. added to a plate of a thickness of about $^1/_2$ in. would strengthen it considerably; but when the same amount is added to a plate 1 in. or more thick, the strength is increased but little.

That this is true is shown by the head rupture described by S. W. Miller in Mechanical Engineering for August, 1926, the stresses at the edge of the manhole being excessive even at the working pressure. It should also be remembered that there has been considerable adverse criticism recently by German and Swiss engineers of the common design of convex heads, that is, those that are spherical with a small knuckle of circular cross-section connecting the main part of the head to the flange.

In what follows the author presents what are believed to be rational methods of design of heads, considering the stresses in all parts of heads concave to the pressure.

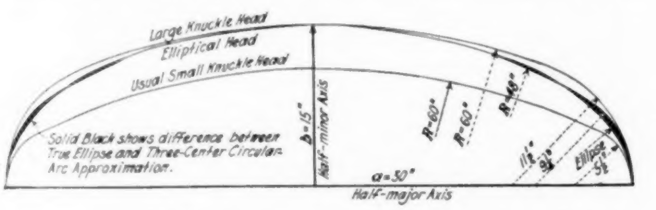


Fig. 1 Shapes of Dished Heads

The first consideration is the shape of the head. In practice it is usual to design the crown of the head with a radius equal to the diameter of the shell, and to use a knuckle of comparatively short radius (Fig. 1). This gives an abrupt change in the radius of curvature of the head at the junction of the knuckle and the crown, and very high bending stresses are set up at this point under conditions of internal pressure. The elastic limit of the material may even be exceeded, and then the head takes on a new shape in which the knuckle merges into the crown gradually, with a steady change in the radius of curvature. If the stress be great enough, the head takes the shape of an ellipsoid. This has been shown experimentally by Bach.[2] The stress in such a head may be calculated as shown in the following paragraphs.

Calculation of Stresses in the Head of a Pressure Vessel

Various attempts have been made to evaluate these stresses, and in each case it has been necessary to make assumptions. After reviewing the literature and attempting several original solutions, the analysis given by Huggenberger[3] has been chosen as the most satisfactory. The assumption on which he bases his calculation is that the cross-section of the head through the axis is an ellipse. Experiments show that if we have a head which is basket-shaped and subjected to internal pressure, bending moments are set up in the head, particularly at the junction of the knuckle with the crown. The head takes the shape of an ellipse. The calculation of the bending moment which causes the head to deform is extremely difficult. Once the shape is assumed, the bending moments are reduced so as to be practically negligible, and the stresses in the head are exactly what we expect in an ellipsoid of revolution.

In a sphere we have the maximum volume for a given surface, and the sphere would be the ideal theoretical shape for the head of a pressure vessel. Due to practical considerations, however, the depth of the dish of the head should be as low as is consistent with theory. The solid of revolution which has maximum volume for a given surface when the depth of the dish is *less* than the radius of the shell is an ellipsoid of revolution. It is clear that the plane tangent to the surface at the point of contact with the circular cross-section of the vessel must be normal to the plane of this cross-section. This condition is also fulfilled by an ellipsoid. Thus the assumption of an ellipsoid is quite rational.

Starting with the stress-strain relations in a solid of revolution and assuming that the head is an ellipsoid of revolution, Huggenberger arrives at the following equations.

$$T_r = \frac{1}{2}[(Rk)^2 + x^2(1-k^2)]^{1/2}\frac{p}{t}$$

$$T_c = 2T_r\left(1 - \frac{1}{2}\,\frac{R^2}{R^2 + x^2\left(\frac{1}{k^2}-1\right)}\right)$$

where T_r = radial tension
T_c = hoop tension or compression
R, a = radius of the cylindrical shell ($^1/_2$ the major axis)
b = semi-minor axis
k = ratio of the major to the minor axis = a/b
x = distance from the minor axis to any point (abscissa)
p = internal pressure
t = thickness of the plate.

Expressed in terms of a and b these equations become

$$T_r = \frac{p}{2t}\left(\frac{a^4 + b^2x^2 - a^2x^2}{b^2}\right)^{1/2}$$

$$T_c = \frac{p}{2t}\left(\frac{a^4 + b^2x^2 - a^2x^2}{b^2}\right)^{1/2}\left(\frac{a^4 + 2b^2x^2 - 2a^2x^2}{a^4 + b^2x^2 - a^2x^2}\right)$$

Let us consider these equations for various special cases. The relation of the radius of the shell to the depth of dish, a/b, is assumed for each particular case. If $a/b = 1.42$, the value of T_c goes from a maximum tension at the center of the crown to zero at the boundary of the knuckle and the cylinder with increasing value of x. If $a/b = 2$, the maximum stress in the crown is tension and is located in the center of the crown; the maximum stress in the knuckle is compression and is located at the boundary of the cylinder and the knuckle. In this case the tension in the crown and compression in the knuckle are of equal magnitude. If a/b is greater than 2, the greatest stress in the crown is still tension and lies at the center, but is far exceeded in magnitude by the compression in the knuckle at the boundary of the knuckle and the cylinder. If the present basket-shaped head is still retained, the radius of the knuckle should be so designed that the perpendicular distance from the center of the head to the base of the knuckle is at least one-half of the radius of the shell.

It is assumed in this discussion that the thickness of the plate in the head is the same as the thickness of the plate in the shell, that is, the thickness of the plate in the head is the same as that of a whole sphere with radius equal to the diameter of the shell. The curves of Fig. 2 show the ratio of stresses at any point on the surface to the stress in the truly spherical head of radius equal to shell diameter. The theoretical shape of the head is an ellipse, and as far as we know

[1] Union Carbide and Carbon Research Laboratories, Inc.
[2] *Zeitschrift des Vereines deutscher Ingenieure*, 1924.
[3] *Ibid.*, vol. 69, no. 6, Feb. 7, 1925, pp. 159–162.

Presented at a joint meeting of the Metropolitan Section and Machine Shop Practice Division of the A.S.M.E. with the New York Section of the American Welding Society, New York, N. Y., January 4, 1927.

there is no practical objection to the manufacture of an elliptical head. However, in case the basket-shaped head is still used, the value of b for substitution in the above formula may be obtained in the following expression derived from the simple trigonometry of the figure:

$$b = 4a \sin \frac{\alpha}{2} + r \cos \alpha \qquad \sin \alpha = \frac{a - r}{2a - r} \ (r = \text{knuckle radius})$$

In conclusion, it should be noted that in cases where a/b is greater than 2 and the thickness of the head is the same as the thickness of the shell, the hoop compression in the knuckle greatly exceeds the maximum fiber stress in the shell. The tangential stress at the center of the head equals the radial stress at the center of the head in any case. For safe construction, the knuckle radius should be so chosen that the ratio of major to minor axis is equal to or less than 2, i.e., the depth of the dish is equal to or greater than $1/4$ of the diameter of the shell. For best construction a true ellipse should be used.

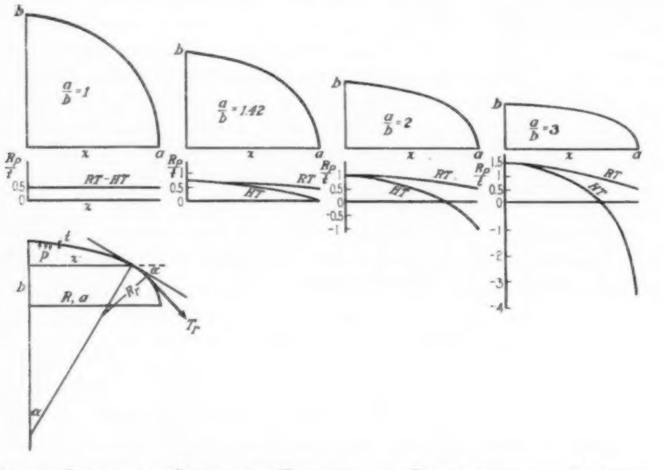


Fig. 2 Ratios of Stress in Ellipsoid of Revolution to Stress in Cylinder with Radius Equal to One-Half the Major Axis a, Versus Distance for Axis of Revolution

(a = radius of shell; b = depth of dish of head; RT = radial tension; HT = hoop tension or compression.)

It should be noted that these calculations assume equal distribution of stresses across the thickness of the plate and also neglect bending moments and compressive forces due to the cylindrical shell at the junction of the shell with the head. That these assumptions are safe has been shown in the experimental work of Bach[2] and Höhn.[4]

Bending Moments in a Non-Spherical Head

Tests indicate that in general a head shaped like a basket, that is, constructed with a large crown radius and small knuckle radius, does not act as a sphere, and that tangential stress only equals the radial stress at the crest of the crown. In fact, in most cases the head acts more like a dished plate supported at the rim than like a section of a sphere. This means that the stresses calculated by the common formula are very much increased, due to bending moments, which can not be readily determined. By constructing the head as an ellipsoid, as indicated in the previous paragraphs, we can approximate the stresses at any point.

Design of a Boiler Head with Manhole

In the case where the head carries a manhole, as in any other member which contains a hole and is subjected to strain, the stresses at the edge of the hole are much greater than they would be in a continuous plate. In the case of a freely supported infinitely large flat circular plate with a finite hole and normal pressure, the fiber stress at the edge of the hole is twice as great as in the rest of the plate. The manner of distribution of these stresses in the plate is very difficult to determine analytically. However, the author believes that the following method of designing flanges for a manhole in a spherical head is rational, and tests indicate its correctness.

In an elliptical head there are spherical conditions at the center of the crown; the stresses fall off as we go toward the knuckle. Moreover, the stresses are entirely symmetrical with respect to the center axis, so that spherical conditions may be assumed for a small area about the center of the crown, this area to contain the manhole. The problem of stresses around a manhole in the head of a pressure vessel is difficult to solve by the direct application of the general relations between stress and strain as formulated in the theory of elasticity. This is due to the fact that there are no definite boundary conditions, the effect of the hole at any given distance from its center being unknown. Moreover, such a solution would have, as a resulting design, a flange whose thickness varied progressively from the edge of the hole outward, according to a complex function. The designer has attacked this problem in a practical way by using a plate of increased thickness together with a flange at the manhole.

The logical solution would seem to be to take care of stresses due to the manhole by the flange itself. Thus the plate will not have to carry abnormal stresses. With this in mind the problem resolves itself into one of finding the stress in a thick hollow cylinder integral with the head. This head exerts a tensional force on the cylinder as shown in Fig. 3. The axis of the cylinder is coincident with the radius of the head at the center of the manhole. This eliminates the head from consideration, leaving, in effect, the very simple problem of determining the stresses in a thick hollow cylinder subjected to internal pressure.

It is appreciated that the above assumption is an approximation,

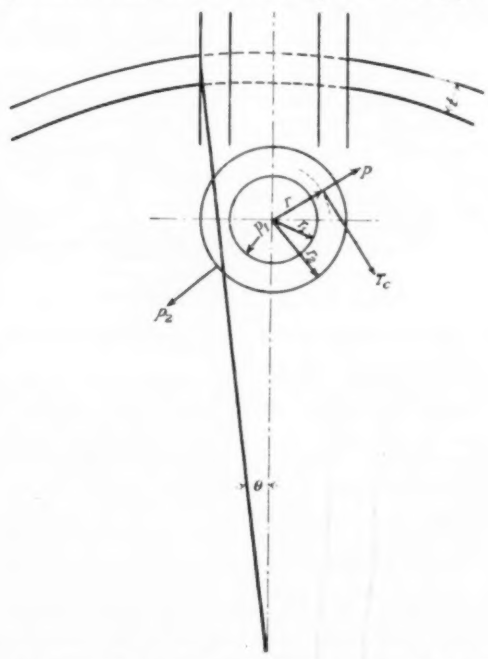


Fig. 3 Tensional Force Exerted on Cylinder by Head

as the stresses immediately adjacent to the inserted cylinder are doubtless changed by the very insertion of this cylinder. In practice, in inserting the cylinder, it is necessary to use fillets to distribute the stress, and it is probable that the mass of metal about the manhole will be greater than theoretically necessary. Thus there is constraint of the head with accompanying increase in stress. This effect has been disregarded and a stress relation deliberately formulated rather than a strain relation. The experimental data show that this procedure is justifiable.

Assuming no manhole whatever, there is a definite tension in the head radial to the axis of the vessel. This will be designated as T_r, and may be readily computed for any particular case. T_r then becomes a tension normal to the surface of the thick hollow cylinder at all points. Thus we have a thick hollow cylinder subjected to an external tension T_r.

The theory of elasticity yields the following formula for the hoop tension at any point of any thick hollow cylinder. This is a modification of the one derived in J. Prescott's Applied Elasticity and is,

[4] *Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, no. 6, Feb. 7, 1925, pp. 155–158.

in effect, Lamé's formula, which gives higher stresses than do others, and is therefore on the safe side.

$$T_c = \frac{1}{r_2^2 - r_1^2}\left\{r_2^2 p_2 + r_1^2 p_1 + \frac{r_1^2 r_2^2}{x^2}(p_1 + p_2)\right\}$$

where T_c = hoop tension at any point with radius x
x = distance of any point in the cylinder from the central axis
r_2 = the outside radius
r_1 = inside radius
p_2 = external pressure (or tension with signs reversed as in formula)
p_1 = internal pressure.

The external tension p_2 will be the T_s of the sphere mentioned in a previous paragraph. Strictly speaking, this value should be multiplied by cos θ, θ being the angle included by the axis and the outside radius of the assumed hollow cylinder at the center of the sphere. However, for practical purposes this cosine may be assumed equal to unity. The smallest-diameter head requiring a manhole according to the A.S.M.E. Unfired Pressure Vessel Code is one of 36 in., and it is readily seen that in this case the error is

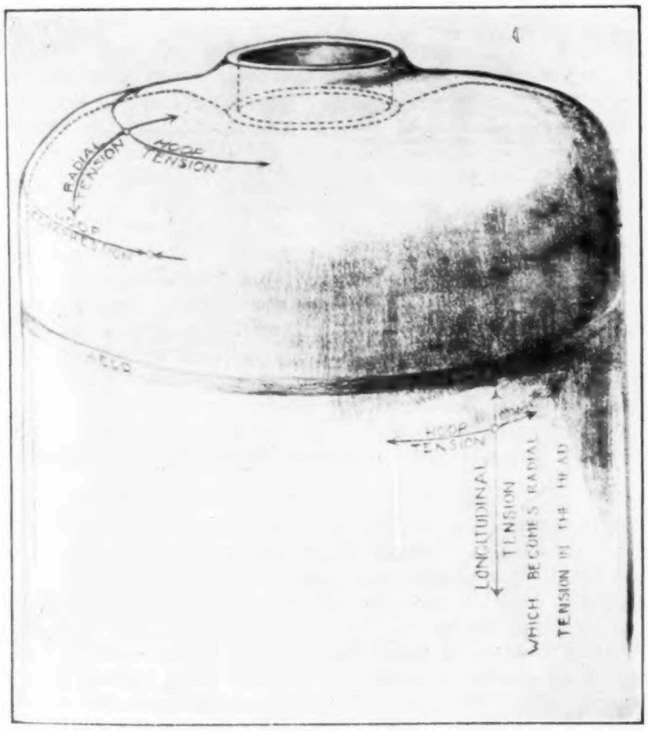


FIG. 4 STRESSES IN HEAD AND SHELL

negligible. The internal pressure is negligible; so $p_1 = 0$. This reduces the formula to:

$$T_c = \frac{p_2}{r_2^2 - r_1^2}\left(r_2^2 + \frac{r_1^2 r_2^2}{x^2}\right)$$

For the stress at the inner surface of the cylinder, $x = r_1$ and

$$T_c = \frac{2r_2^2}{r_2^2 - r_1^2}\,p_2$$

For stress at the outer surface, $x = r_2$ and $T_c = \dfrac{r_2^2 + r_1^2}{r_2^2 - r_1^2}\,p_2$

If the value T_c/p_2 is to be calculated, r_1 being assumed in any particular case and r_2 being expressed as a multiple of r_1, the curves shown in Fig. 5 result. Naturally, as $r_2 - r_1$ approaches zero, that is, as the wall thickness approaches zero, the tension approaches infinity. Where r_2 is infinite, values for T_c/p_2 are 1 and 2 at the outer and inner surfaces, respectively. That is, the curves are asymptotic to r_1 and to 1 and 2. Note that from a theoretical standpoint, using the ordinary stress-strain relations, it is found that in a plate containing a hole of finite size the stress at the inner surface or edge of the hole is twice the stress at an infinite distance from the hole, when the tension is uniform in all directions.

In the design of a flange for a manhole in a head of a thickness t, we can assume any value greater than 2 for $\dfrac{h}{t} = \dfrac{T_c}{p_2}$, h being the length of the cylinder. From this the distance from the center to the point where the effect of the hole extends is determined. Thus the cross-section of the flange is found. As the stress at the outer surface

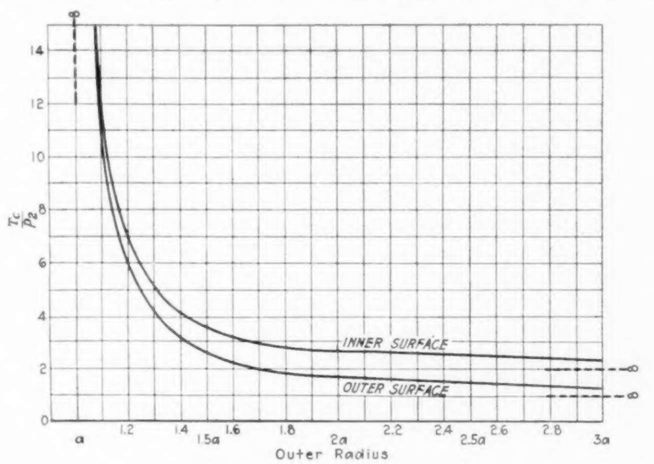


FIG. 5 RATIO OF STRESS AT INNER AND OUTER SURFACES OF A THICK HOLLOW CIRCULAR CYLINDER OF CONSTANT INTERNAL RADIUS, SUBJECTED TO CONSTANT EXTERNAL TENSION, TO THAT TENSION VERSUS VARIABLE OUTER RADIUS EXPRESSED AS A MULTIPLE OF THE INNER RADIUS

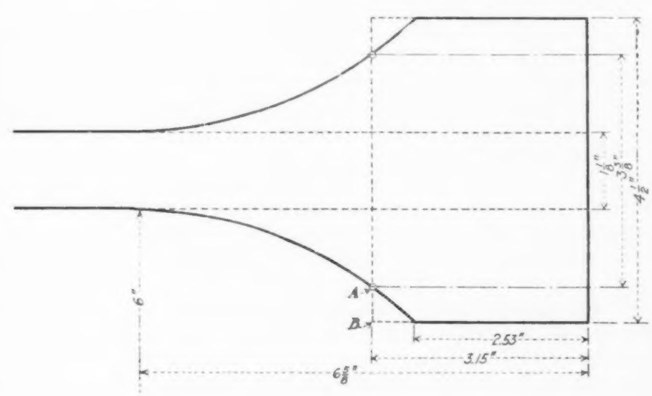


FIG. 6 SECTION OF MANHOLE FLANGE

is less than that at the inner surface, the length of the cylinder at this point may be reduced accordingly. The exact length at any point in the hollow cylinder may be readily calculated from the above formula. From the engineering point of view it is most practicable to make this section rectangular.

The problem of the fillet to be used in joining the head to the thick cylinder next presents itself. The previous computations assume that the stress T_s is uniformly distributed over the outer surface of the cylinder. This is accomplished by the use of a fillet properly proportioned. It will be first assumed that the ability of the material to transfer stress is just such as to appropriately distribute the increase in stress given by the formula on approaching the inner surface. In this case the fillet takes the shape of the outer surface curve shown in Fig. 5, passes through the point on the outer surface of the cylinder corresponding to the curve (Point A Fig. 6) and is asymptotic to the surface of a spherical plate of infinite radius. Now, if the ability of the material to transfer stress is such as to give superior and wider distribution than called for by the first assumption, the section of the hollow cylinder further from the head than any given transverse plane carries some of this stress. Accordingly, at any given distance from the inner surface we may reduce the height of the fillet curve. A practical way in which to do this and one which gives a fillet curve for a reasonable height of flange in accordance with engineering experience is as follows:

Referring to Fig. 6, pass the outer-surface curve through the outside corner B of the computed flange and asymptotic to the head plate. The length of the cylinder at this point being greater by t than necessary at the outer surface, move the fillet curve a distance t to point A. The length at the outer surface will then correspond to the formula, and the fillet curve will cut the head surface at a definite distance from the outer surface of the cylinder. This distance is proportional to the difference between the abscissas of the respective point on the outer surface curve and T_c/p_2.

There are certain refinements applicable to the formulas above used. The general formula is based on the assumptions that the strain in the direction of the axis of the thick hollow cylinder is either zero or constant, and that the displacement perpendicular to the axis is radial and depends only on the radius. In the case of a manhole with an inside cover, we really have a thick cylinder with end pressure. This end pressure has a component in the plane perpendicular to the axis according to Poisson's ratio, which tends to increase the tension in the thick hollow cylinder. This quantity, however, is comparatively small, and we may apply the ordinary formula for end stress on a cylinder, $T_c = pr/2t$. In this case t is the thickness of the hollow cylinder and $2t$ is of the same general order as the radius, so that the tension is fairly low. It may be computed for any particular case, and if sufficiently great, allowance should be made.

It has been pointed out by Professor Beyer of Columbia University that the cover of the manhole bearing on the end of the thick cylinder will also give a stress normal to the axis of the cylinder due to arch effect. This stress will be transmitted through the gasket to the cylinder. The thickness of the plate is generally sufficient so that this force is negligible. However, should the cover be so designed and held in place that this force becomes appreciable, it must be taken into consideration.

There is also a pressure on that section of the hollow cylinder extending into the shell. This pressure tends to counteract the above-mentioned end effects. The compressive force or negative tension due to this outside pressure may be readily computed by the ordinary formula, $T = pr/t$, and in any particular case the value of T may be calculated and subtracted from the value T_c obtained by the general formula.

Professor Krefeld of Columbia University has noted that in a flange resulting from the above design the radial stress has a normal component which tends to counteract the tangential stress, so that strain-gage measurements will only give effective stress for the usual value of the modulus.

In the case of elliptical manholes the greatest tension occurs at the end of the major axis. Calculations for a circular manhole may be directly applied, using the major axis as the diameter. The use of the usual elliptical manhole means that there will be bending moments set up in the flange. However, the moment of resistance of the flange is so great compared to its diameter that these bending moments are not serious and probably can be neglected.

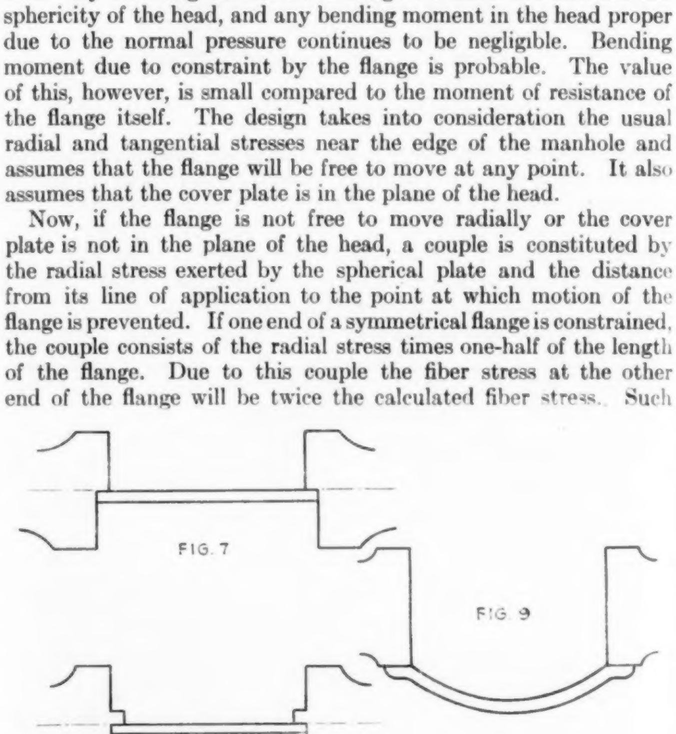
In the case of a manhole with outside cover the stresses are somewhat different. In effect we have a new problem. The terms containing p_1 in the general formula now take on a real value, which is additive instead of negligible, and the end effect is tensional, and therefore negative. In the actual design of a manhole with outside cover, the metal removed in order to insert studs must also be taken into consideration.

As an example of the application of these formulas, consider a pressure-vessel head with a manhole 15 in. in diameter at the center, the manhole having an inside cover. Assume the working pressure to be 300 lb. per sq. in., and the maximum allowable working stress, 9000 lb. per sq. in. The head without the manhole is taken as $1^1/_8$ in. thick. The choice of the height of flange or length of cylinder must first be made. Taking a flange having four times the thickness of the plate, i.e., $h = 4^1/_2$ in. $h/t = 4 = T_c/p_2$. From the curves in Fig. 2, $T_c/p_2 = 4$ at the inner surface, $r_2 = 1.4r_1$; r_1 is specified as $7^1/_2$ in., therefore $r_2 = 10.65$ in. and the flange thickness is 3.15 in. Now T_c/p_2 at the outer surface of the cylinder is 3, so that $h = 3^3/_8$ in. But for $T_c/p_2 = 4$, $r_2 = 1.3\ r_1$; and for $T_c/p_2 = 2$, $r_2 = 1.7\ r_1$. The width of fillet is $0.4\ r_1 = 3$ in. and distance from inner surface to end of fillet is 6.15 in. The curve connecting these points should be the shape of the outer-surface curve. This will cut the corner from the originally calculated flange so that the final dimensions of the rectangular section are $4^1/_2 \times 2.53$ in. Fig. 3 shows the design. It will be noted that a radius has been substituted for the calculated fillet curve and the curvature of the head neglected.

Bending Moment in Manhole Flanges

It is well known that in a sphere subjected to internal pressure, the pressure is normal to the surface and the bending moment is zero. By inserting the manhole flange we have not affected the sphericity of the head, and any bending moment in the head proper due to the normal pressure continues to be negligible. Bending moment due to constraint by the flange is probable. The value of this, however, is small compared to the moment of resistance of the flange itself. The design takes into consideration the usual radial and tangential stresses near the edge of the manhole and assumes that the flange will be free to move at any point. It also assumes that the cover plate is in the plane of the head.

Now, if the flange is not free to move radially or the cover plate is not in the plane of the head, a couple is constituted by the radial stress exerted by the spherical plate and the distance from its line of application to the point at which motion of the flange is prevented. If one end of a symmetrical flange is constrained, the couple consists of the radial stress times one-half of the length of the flange. Due to this couple the fiber stress at the other end of the flange will be twice the calculated fiber stress. Such constraint may be caused by the cover plate fastened on the inner end of the flange. Professor Beyer has pointed out how this plate may act either as an arch or as a tension member.



Figs. 7–9 Three Methods of Avoiding Bending Moment in Manhole Flanges

Several methods may be used to overcome this. The couple may be eliminated by reducing its arm to zero. Figs. 7, 8, and 9 illustrate methods of doing this. In Fig. 8 the cover plate rests against an annular shoulder, the shoulder being so fixed that the bearing surface coincides with the center surface of the spherical head. Note that this means a manhole with a radius 1 in. greater than ordinarily used. In Fig. 7 the same general scheme is applied, but instead of using an annular shoulder the upper section of the flange proper acts as a shoulder and may be calculated on the original radius, the lower section being calculated on a radius 1 in. greater. In each of these cases the arm of the couple is zero. Fig. 9 shows a method of eliminating any tendency to constrain the radial motion of the flange, the cover plate being so designed that the perimeter moves in the same manner as the flange proper. This of course is very difficult to accomplish. The method illustrated in Fig. 7 is correct from the theoretical point of view, regardless of the manner in which the plate is fastened.

Conclusion

In conclusion, the author would state that experiments and calculations both in Europe and in this country show that:

1 The present method of designing pressure-vessel heads is subject to improvement.

2 The design herein described, namely, an ellipsoidal head with a ratio of axes of 2 to 1, is much more satisfactory.

3 The manhole flange symmetrically placed and calculated as a hollow cylinder is also much more satisfactory.

APPENDIX NO. 1

Calculation of the Plate Thickness of Dished Heads with Knuckles According to the Hamburg Rules[1]

BY A. HUGGENBERGER,[2] ZURICH

The stress conditions in elliptical shells are established from theoretical and experimental analyses. On the basis of the relation between the radius of curvature and maximum stresses, the danger of rupture in small radius boiler knuckles is approximately determined. The calculation of the thickness of the plate according to the Hamburg rules is discussed and it is shown that the use of too small a value for the radius of the knuckle will give too small a value for the thickness of the plate. This is explained with reference to the tensile strength of the material and to an insufficient head tension with insufficient radius of curvature of the knuckle. Thus the appropriate head tension can be so shown that the usual calculations will give the correct thickness of the plate.

THE statistics as to boiler failures show that the dangerous accidents occur due to bursting in the knuckle. If we compare the plate thickness in the knuckle with the usual formula thickness given by the Hamburg rules, we find that in many cases the plate in question is thick enough and that the reason for the failure lies in too small a knuckle radius. The knuckle radius is not taken into consideration in the Hamburg rules. That this measurement is particularly important in considering the ultimate strength of the head has already been shown by Bach. The knuckle stresses determine the ultimate strength of the head in the case of small knuckle radii. To date, there are available only very rough figures taken from the displacement of the crown and the knuckle to indicate the size of the stresses. Figures on heads of various knuckle radii show that the knuckle radius can be many times the crown stress.

From the investigation of the ultimate strength of balls and hollow cylinders which is set forth in the theoretical work of Prof. Dr. Meiszner, one can calculate exactly the strength of the head. Bending moments, tensile stress, and compression which are carried into the head from the cylindrical shell must be evaluated as boundary conditions. The Society for the Inspection of Steam Boilers for which this work was originally carried on has never applied the results of these calculations nor ever prescribed them.

STRESS CONDITIONS IN ELLIPTICAL SHELLS

The head can be calculated according to the theory of thin-walled containers when the plate thickness is small in comparison with the other dimensions and shear and bending moment can be neglected. The following equations are satisfied by both main stresses, σ_m and σ_u which distribute themselves equally over the cross-section:

$$\frac{\sigma_m}{R_m} + \frac{\sigma_u}{R_u} = \frac{p}{s} \qquad [1a]$$

$$\sigma_m = \frac{r}{\sin\alpha}\,\frac{1}{2s}\,p \qquad [1b]$$

$$R_m = \frac{(1 + z'^2)^{3/2}}{z''} \qquad [2a]$$

$$R_u = \frac{r}{\sin\alpha} \qquad [2b]$$

Referring to Fig. 1:

p = normal pressure at any point in the container (kg. per sq. cm.)
s = constant wall thickness (cm.)
P = any point in the central cross-sectional curve
r, z = rectangular coördinates of P
α = angle between the normal to the surface $\overline{PS} = n$ and the axis of revolution $\overline{MS}$
z', z'' = derivatives of the function $z = f(r)$ with respect to r
σ_m = principal stress in the direction of the tangent to the cross-sectional curve (radial tension, kg. per sq. cm.)
σ_u = principal stress in the direction of the tangent to the small circle (hoop tension, kg. per sq. cm.)
σ_B = allowable stress in the crown of the head (650 kg. per sq. cm. for boiler plate according to the Hamburg rules)
R_m = radius of curvature at the point P of the cross-sectional curve (cm.)
ρ = radius of curvature of the cross-sectional curve at the knuckle, Fig. 2 (cm.)
R = radius of curvature in the head M (cm.)
R_u = radius of curvature at the point P in the plane through the line normal to the surface n, and normal to the cross-sectional plane

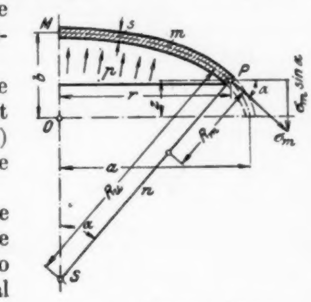


FIG. 1 SYMBOLS USED

R_1 = radius of curvature in the crown of the inner surface of the shell (cm.)
R_2 = radius of curvature in the knuckle of the inner surface of the shell (cm.)
a, b = major and minor semi-axes of the elliptical head (cm.)
k = a/b = ratio of major to minor axis.

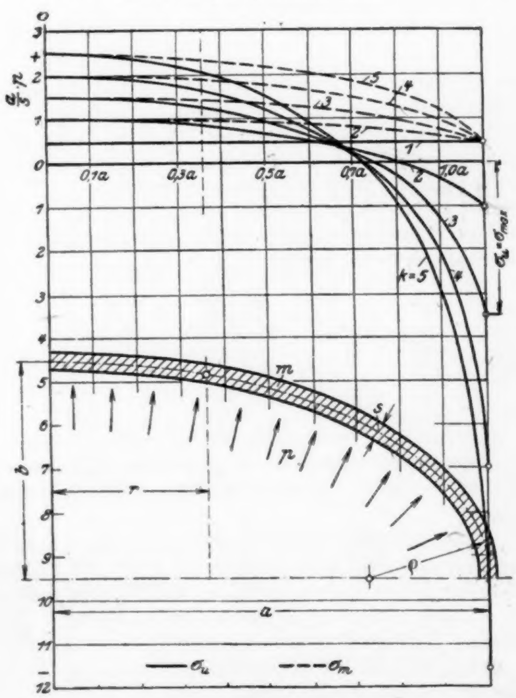


FIG. 2 STRESSES σ_u AND σ_m IN ELLIPTICAL HEADS WITH VARYING RADIUS r

If the equation of the cross-sectional curve of the head is known, the principal stresses σ_m and σ_u can be evaluated by means of formulas [1a] and [2b]. Experiments show that the curve of the usual basket-shaped cross-section of boiler head approaches an ellipse under increasing internal pressure. The equation of the ellipse of Fig. 1 is

$$\frac{r^2}{a^2} + \frac{z^2}{b^2} = 1 \qquad [3]$$

From Equation [3] and $b = a/k$,

$$\sigma_m = \frac{1}{2}\left[(ak)^2 + r^2(1-k^2)\right]^{1/2} \times \frac{p}{s} \qquad [4a]$$

$$\sigma_u = \left[(ak)^2 + r^2(1-k^2)\right]^{1/2}\left[1 - \frac{1}{2}\,\frac{a^2}{a^2 + r^2(1/k^2 - 1)}\right]\frac{p}{s} \qquad [4b]$$

[1] Translated from *Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, no. 6, February 7, 1925, pp. 159–162.

[2] Engineer, Swiss Association of Steam-Boiler Owners.

If we eliminate k in these equations and put $r = a$ and $b = \infty$, we have the following expressions for the tension in the cylindrical shell:

$$\sigma_m = \frac{a}{2s} p \quad [5a]$$

$$\sigma_u = \frac{a}{s} p \quad [5b]$$

Equations [4a] and [4b] have the same meaning with respect to the calculation of the plate thickness of elliptical heads as has the boiler formula for the calculation for the plate thickness of cylindrical shells. The assumption that $\sigma_u = 0$ gives from Equation [4b], the radius r of the small circle for which the hoop tension disappears. For larger values of r, σ_u becomes negative.

$$\sigma_u = 0, \; r = \frac{a}{\sqrt{2(1 - 1/k^2)}} \quad [6]$$

For the circumference of the shell, $r = a$, we have

$$\sigma_m = \frac{a}{2s} p \quad [7a]$$

$$\sigma_u = \frac{a}{s}\left(1 - \frac{1}{2}k^2\right) p \; (\sigma_{\text{max.}} \text{ for } k > 2) \quad [7b]$$

The radial tension at the circumference of the shell is independent of the ratio of the elliptical axes. In the crown of the head the two principal stresses are equal, namely:

$$r = 0, \; \sigma_m = \sigma_u = \frac{ak}{2s} p \; (\sigma_{\text{max.}} \text{ for } k < 2) \quad [8]$$

The plate thickness of the elliptical head, neglecting compression and bending at the edge, can be expressed as a function of k, depending upon whether k is greater or less than 2, Equation [7b] or [8]. In the calculation using Equation [8], σ_u, the compressive stress in the knuckle, must be taken as negative.

With $ak = a^2/b = R$ as radius of curvature of the crown of the elliptical shell, Equation [8] takes the following forms for the plate thickness or the internal pressure:

$$s = \frac{R}{2\sigma_B} p \quad [9]$$

$$p = \frac{2s\sigma_B}{R} \quad [10]$$

In these equations $\sigma_m = \sigma_u = \sigma_B$ are the allowable stresses. Equations [9] and [10] are the basis of the Hamburg rules for the calculation of the plate thickness and the allowable pressure on dished heads.

In Fig. 2 the principal stresses σ_m and σ_u are given according to Equations [4a] and [4b] as a function of r for the elliptical-axis ratios of 1 to 5. If $k = 1$, then $a = b$, i.e., the elliptical shell becomes a sphere. In this case R in Equations [9] and [10] is the radius of the sphere. Both stresses are equal for all points on the spherical surface. For $k = 1.42$ the positive hoop tension σ_u at the circumference of the head becomes zero. As k increases σ_u becomes negative. As long as k is less than 2 the greatest stress takes place in the crown. For $k = 2$ the hoop stress in the knuckle is equal to the tension in the crown. For k greater than 2 the greatest stress ($\sigma_u = \sigma_{\text{max.}}$, negative hoop tension) is at the circumference of the shell ($r = a$).

Increasing flattening out of the head (increasing k) produces a greater stress. As $\rho = \frac{b^2}{a} = \frac{a}{k^2}$ is the radius of curvature at the end of the major axis, the knuckle stress increases rapidly with decrease in the radius of curvature of the knuckle. The effect of small knuckle radii is clearly shown in Fig. 3 by the steep increase of σ_u' (value of σ_u for $r = a$). In order to avoid high stresses in the knuckle a knuckle radius as large as possible must be used. This holds for elliptical as well as basket-shaped heads.

As it is impractical to press a deep head, for example, one where k is less than 2, the axis ratio $k = a/b = 2$ (Fig. 3, dotted ordinate) is taken as the most practical and at the same time sufficiently strong head. This ratio implies a knuckle radius equal to $0.25a$. In many of the heads now in use this radius is roughly $0.1a$. According to Fig. 3 the radius of the knuckle of the elliptical head there shown was four times as great as for $\rho = 0.25a$. From observations on heads with smaller knuckle radii, Bach has come to the same conclusion. If ρ is less than $0.1a$ this ratio becomes even higher, so that the knuckle stress in the case of small knuckle radii

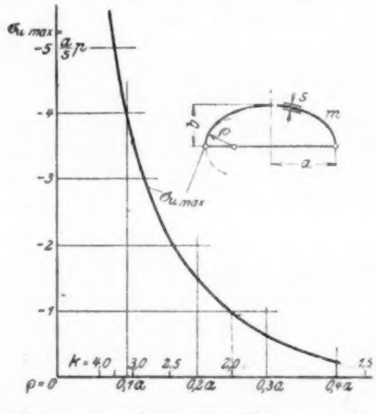
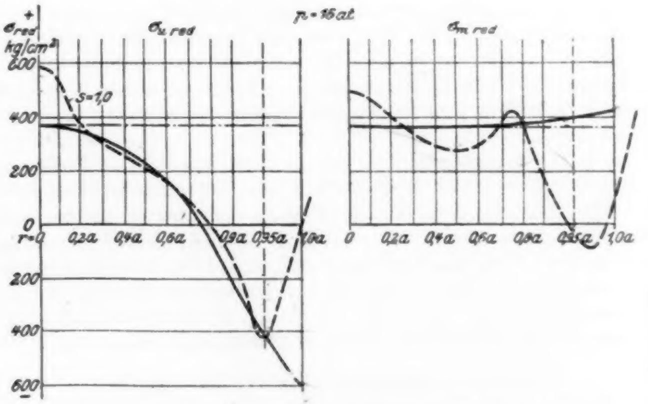


FIG. 3 KNUCKLE STRESS σ_u IN ELLIPTICAL HEAD WITH VARYING KNUCKLE RADIUS



FIGS. 4 AND 5 REDUCED STRESSES IN ELLIPTICAL HEAD NO. 8 WITH 235 LB. PER SQ. IN. (16 ATMOS.) PRESSURE

increases to an inadmissable value, even though the plate thickness for other points is sufficiently great.

REDUCED STRESSES IN A CONTAINER WITH AN ELLIPTICAL HEAD

Among other things the author has measured the stresses of a container with an elliptical head ($k = 1.97$) and one with a basket-shaped head in which $k = 4.12$. The measurements are shown in Table 1. The maximum allowable pressure according to the Hamburg rules (Equation [10], $\sigma_B = 650$ kg. per sq. cm.) is 20 atmospheres. In order to compare theory and practice the author presents Figs. 4 and 5 in which the reduced stresses in the elliptical head are shown dotted and are taken from Figs. 7 and 8 of Mr. Höhn's article; also the stresses obtained from Equations [4a] and [4b] and the relations

Poisson's ratio

$$\sigma_{m\,\text{red}} = \sigma_m - r\,\sigma_u \quad [11a]$$

$$\sigma_{u\,\text{red}} = \sigma_u - r\,\sigma_m \quad [11b]$$

which give the stress in an elliptical head without the effect of the cylindrical shell.

TABLE 1 DATA ON DISHED HEADS (DIMENSIONS IN CENTIMETERS)

No. of container	Cross-section	s	a	b	$k = a/b$	$d = 2a - s$	R	$R_1 = R - 0.5s$	ρ	ρ/a	$R_2 = \rho - 0.5s$
8	Elliptical	1.0–1.2	39.4	20.0	1.97	77.6	77.5	76.9	10.1	0.268	9.5
7	Basket-shaped	1.2	59.3	14.4	4.12	117.4	160.6	160.0	3.9	0.066	3.3

TABLE 2 PRINCIPAL MEASURED AND CALCULATED VALUES OF REDUCED AND UNREDUCED STRESSES (IN KG. PER SQ. CM.) FOR p = 8 ATMOSPHERES

(Values in bold-face type calculated from Equations [2], [4], [7], and [11])

Form of Head	r/a	r, cm.	α, deg.	$\sigma_{m\ red}$	$\sigma_{u\ red}$	σ_m	σ_u	s, cm.	Remarks
Elliptical, a = 39.4 cm.; plate thickness in crown, s = 1.0 cm. (Container No. 8)	0	0	0	+ 230	+230	+ 329	+ 329	1.0	Crown of head: max. stress, $\sigma_{max.}$
				+ **217**	+**217**	+ **310**	+ **330**		
				....	...	+ 274	+ 274	1.2	
				+ **181**	+**181**	+ **258**	+ **258**		
	0.95	37.4	60	− 40	−200	− 110	− 233	1.2	Knuckle
				− **195**	−**198**	+ **149**	− **153**		
	1.06	39.4	90	+ 40	0	+ 44	+ 13	1.2	Rim of head
				+ **205**	−**286**	+ **131**	− **247**		
Basket shape, a = 39.3 cm. (Container No. 7)	0	0	0	+ 381	+381	+ 544	+ 544	1.2	Crown of head
	0.98	58.2	45	−1080	−860	−1470	−1300	1.2	Knuckle: max. stress. $\sigma_{max.}$ (stress in section)
	1.0	59.3	90	− 930	−900	−1320	−1295	1.2	Rim of head

The increase of the stress in the crown of the head becomes apparent, as there the plate thickness is just 1 cm. In the remainder of the head the relation of the reduced stress in the head and shell approximates practice. Only in the region of the knuckle are there appreciable differences which are due to the effect of the cylindrical shell. The circumference of the shell has a tendency to draw together, so that the reduced hoop stress is understandable. ($\sigma_{u\ red}$ = 286 kg. per sq. cm.) As the cylinder expands under increased pressure p, the attached knuckle of the head expands and the reduced hoop compression falls off. ($\sigma'_{u\ red}$ = 200 kg. per sq. cm.) Because of the action of the cylinder the tension at the circumference of the head is less than in the case of the head without the cylinder, i.e., k can be increased to 2.3 without increasing the maximum stress in the knuckle beyond the crown stress.

In a similar manner the difference between the theoretical and the measured radial force can also be explained. The shell has a tendency to pull itself together so that there is an outside tension ($\sigma_{m\ red}$ = + 205 kg. per sq. cm.). Through the action of the cylinder the knuckle is extended, i.e., the reduced stress is smaller than in the shell and can become negative ($\sigma'_{m\ red}$ = — 40 kg. per sq. cm.). For very small knuckle radii this effect is even more pronounced.

Table 2 shows the agreement with theory of the maximum stress in the crown of the elliptical head. According to the Hamburg rules the allowable stress in the head is chosen as small as possible, namely, 650 kg. per sq. cm., so that the stress at other points will not be greater than the maximum allowable stress in the material, or 1200 kg. per sq. cm. As the crown tension in these heads with knuckle radii ρ = 0.268 a is the maximum stress in the head, we can increase our head tension to this maximum allowable value. According to Equation [9], for a pressure of 20 atmospheres a plate thickness of 0.65 cm. will suffice, and to use 1.2-cm. plate would be wasteful.

If the knuckle radius is very small, for example, 0.06a, the calculations give a plate which is not sufficiently strong at all points. We have in the case of container No. 7 with the usual basket form cross-sectional measurements of the expansion. With the measurements shown in Table 1 this head, according to Equation [10], with σ_B = 650 kg. per sq. cm., allows a working pressure p of 9.7 atmospheres.

That a radial tension in the case of very small knuckle radii in view of the action of the cylinder can become particularly high is shown in Table 2. With p = 8 atmospheres, the greatest radial stress in the knuckle of the basket-shaped head No. 7 was —1470 kg. per sq. cm. at the knuckle, while in the head plate +544 kg. per sq. cm. was measured. The ratio of the maximum stress in the knuckle to that in the crown is 2.7, while in the case of the elliptical head it is 0.85. For the maximum allowable pressure p = 9.7 atmospheres, which the crown tension of 650 kg. per sq. cm. determines, we have in the case of the basket-shaped head a maximum knuckle stress of 1750 kg. per sq. cm., which is not safely permissible.

According to Equation [9], the plate thickness calculated according to the Hamburg rules is unnecessarily thick in the case of large knuckle radii, and not sufficiently thick in the case of small knuckle radii.

Selection of Allowable Stresses with Respect to The Knuckle Radius

The usual calculation for the thickness of the head according to the Hamburg rules is based on Equation [9] ($s = Rp/2\sigma_B$). The equation for the stress in the spherical shell as well as the equation for the stress in the cylindrical shell can be developed from the equations for the elliptical shell [4a] and [4b], when $a = b = R$, and $r^2 + z^2 = a^2$. The dimensions of the head determine the radius of curvature of the crown. Aside from the above-mentioned rim size of the head, the dimensions of the head, particularly the radius of curvature of the knuckle—which determines the shape of the cross-sectional curve—are not considered. Figs. 4 and 5 give the reduced stresses in the spherical shell with crown radius R = 77.5 cm. equal to that of the elliptical head. These show plainly that Equation [10] holds only for the center point of the crown, and even then only when the center portion is spherical right up to the knuckle.

So long as the maximum stress in the head is in the crown, the formula for spherical shells can be used without objection for

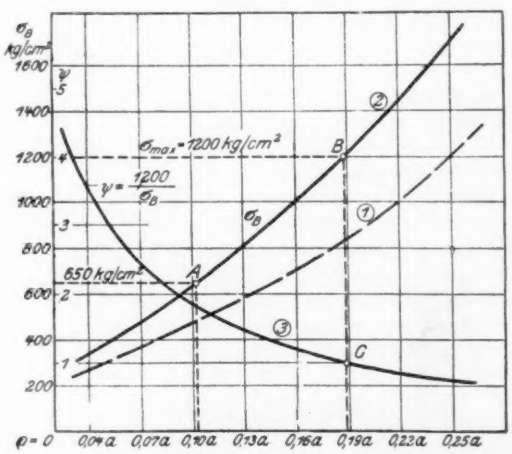


FIG. 6 ALLOWABLE STRESS IN CROWN WITH MATERIAL AT 1200 KG. PER SQ. CM. (17,000 LB. PER SQ. IN.) WITH VARYING KNUCKLE RADIUS

the determination of the plate thickness. The allowable stress σ_B can be chosen much higher, for example, 1200 kg. per sq. cm. However, if the maximum stress does not occur in the crown but in the knuckle, as is the case with small knuckle radii, the entire spherical shell theory is inapplicable.

The Hamburg rules do not endeavor to avoid stresses in the knuckle greater than in the crown, but to keep them as low as possible by limiting the fiber stresses in the crown to a value of 650 kg. per sq. cm.

The unsatisfactory nature of this method of calculation can be seen at once if we calculate s according to a formula which considers the outside radius and the maximum allowable material fiber stress, as well as the remaining measurements $a + \rho$ and the boundary conditions. As such an equation would be too complicated for practical application to boiler construction, the usual formulas should be adapted to the results here shown.

For a shell without cylindrical boundary conditions the maximum stress is determined by the radius of the knuckle. We can therefore ask how great the stress (σ_B) in the crown may be permitted to become for a given knuckle radius so that in no section of the shell, knuckle included, shall the maximum stress be more than 1200 kg. per cm.²—for example, a predetermined value.

The relation of the fiber stress in the crown to the maximum knuckle stress becomes for the ellipsoid of revolution, according to Formulas [7] and [8], as follows:

$$k = \frac{a}{b} > 2 \quad \text{and } \rho = \frac{b^2}{a}$$

$$\frac{\sigma_B}{\sigma_{\max.}} = \frac{\sqrt{a\rho}}{2\rho - a} \quad \ldots\ldots\ldots\ldots\ldots\ldots \quad [12]$$

From these equations the allowable stress in the crown with a given knuckle radius and shell diameter $2a$ is determined. The values for a maximum fiber stress of 1200 kg. per sq. cm. are shown in curve 1 of Fig. 6.

In the case of a head with cylindrical boundary conditions, the state of the stress very close to the knuckle is affected by the bending moment and shearing forces which the cylindrical shell impresses as boundary conditions. From the experiments it is seen that the greatest knuckle stress is not at the circumference of the head. For the elliptical head, Fig. [4], with $\rho/a = 0.268$, the maximum knuckle stress $\sigma_{\max.}$ lies in the small circle $r = 0.95a$. In the case of basket-shaped heads with $\rho/a = 0.066$, $r = 0.98a$. The location of the maximum knuckle stress approaches the periphery of the head as the knuckle radius decreases for constant shell diameters.

In applying these results we selected a predetermined value r/a such that the largest knuckle radius is of reasonable size, and determine the stress according to Equation [4*b*]. This new value is taken as an approximation of the maximum knuckle stress, and Equation [12] is established in a similar manner. The relation between the knuckle radius and permissible stress is shown in curve 2 of Fig. 6. For knuckle radii less than $0.11a$ we must choose a value for our fiber stress less than 650 kg. per sq. cm. For knuckle radii greater than $0.19a$, corresponding to a ratio $k = 2.3$, the maximum stress no longer occurs in the knuckle, but in the crown of the elliptical head. With the help of the experimentally determined relations between ρ, a, and σ_B, we can apply Equation [9] to the stress conditions of the head. The applicability of σ_B in Fig. 6 should be further investigated for the case of the basket-shaped head.

In this theoretical analysis it has been assumed that the stresses over the cross-section (thickness of the plate) are equally distributed. This condition is influenced by bending moments and shearing forces at the boundary as in Figs. 4 and 5, so that it is possible that the stresses on the inside and outside surfaces of the head may be different. This question may be further studied by means of the exact theory of hollow containers.

APPENDIX NO. 2

Strains in Dished Heads[1]

By E. HÖHN,[2] ZURICH

Through a number of independent experiments the best form for dished heads was determined. From the values of measured stress conditions on the outside the strain conditions on the inside are calculated.

Determination of Permanent Set and Elastic Deformation

Permanent Deformation. The author has had the opportunity of making pressure tests of the strength of several electrically welded hollow cylindrical vessels. Each of these vessels or containers was 80 cm. in diameter, and had a plate thickness of 5–9 mm. Each had an electrically welded longitudinal seam as well as two electrically welded heads of basket-shaped cross-section such as No. 7, Fig. 1. In one case, No. 8 (Fig. 2), the cross-section of the head was elliptical. Test pressure was increased up to the point of failure. The vessel burst both at the circular and the longitudinal welds, and the plate ruptured in the knuckle. In the case of vessel No. 1 one of the circular welds broke all the way around and the head was shot off in the direction of the main axis. Up to this point the boiler was tight.

Great care was taken in observing the condition of the head during the pressure tests. By means of sheet templets the shape before and after the pressure tests was determined.

Fig. 3 shows the cross-section of the head of pressure vessels Nos. 1 and 2 before the pressure test (broken lines) and afterward (full lines), the dimensions being as follows:

Plate thickness, 12 mm. (0.47 in.)
Radius of head, 950 mm. (37.4 in.)
Inside radius of knuckle, 45 mm. (1.77 in.)
Outside diameter of cylindrical end, 798 mm. (3.14 in.).

The permanent set under the action of the test pressure, 74 atmospheres (1090 lb. per sq. in.), was determined. The cross-section was almost exactly an ellipse with one-half the major axis a = 400 mm. (15.7 in.) and half the minor axis b = 192 mm. (7.55 in.), making the ratio of a to b practically 2 to 1. The ellipse is shown

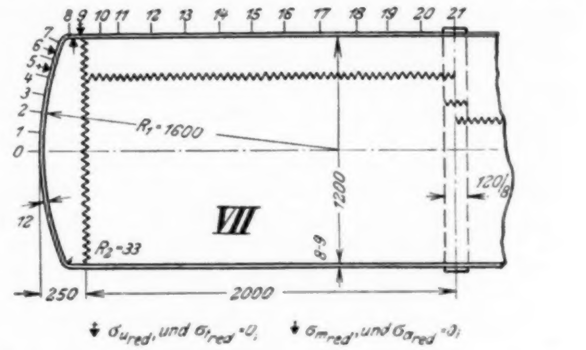


Fig. 1 Electrically Welded Vessel with Basket-Shaped Cross-Section of Head

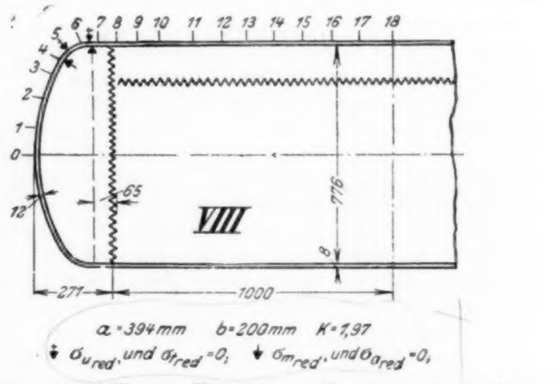


Fig. 2 Electrically Welded Vessel with Elliptical Cross-Section of Head

by a dot-dash line and the difference between the ellipse and the final assumed form is cross-hatched.

Fig. 4 shows the same for the head of vessel No. 7. This was larger than Nos. 1 and 2, its dimensions being:

Outside diameter of cylinder, 1200 mm. (47. 3 in.)
Plate thickness, 12 mm. (0.47 in.)
Head radius, 1600 mm. (63 in.)
Inside knuckle radius, 33 mm. (1.3 in.), or less than for Nos. 1 and 2.

In this case also the shape of the cross-section approximates that of an ellipse. Differences are shown in cross-hatching. They would be even less in the region of the head close to the cylinder if it were not for the effect of the cylinder on the head, the thickness of the head being greater than that of the cylinder. The ellipse has a major semi-axis of 600 mm. (23.6 in.) and a minor semi-axis of 290 mm. (11.4 in.), their ratio being approximately 2 to 1. This permanent set is obtained with a pressure of 59 atmospheres (868 lb. per sq. in.).

In both cases the knuckle spread out and the circular weld was subjected to bending moments. Formerly it was assumed that it was subjected only to tension.

Finally, a test was made on vessel No. 8. This had a head with elliptical cross-section to start with, as the author wished to learn the conditions as compared with heads of basket-shaped cross-section. The dimensions were as follows:

[1] Translated from *Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, no. 6, Feb. 7, 1925, pp. 155–158.
[2] Chief Engineer, Swiss Association of Steam Boiler Owners.

Outside diameter of cylinder, 805 mm. (31.7 in.)
Major semi-axis of ellipse 388 mm. (15.3 in.)
Minor semi-axis of ellipse, 194 mm. (9.62 in.)
Plate thickness, 12 mm. (0.47 in.).

The templet which was made from the head after a pressure of 74 atmospheres (1090 lb. per sq. in.) was applied differed in no way from the templet made before the test. The author believes that this shows conclusively that the ellipse is the natural shape for the cross-section of the head of a hollow cylinder subjected to

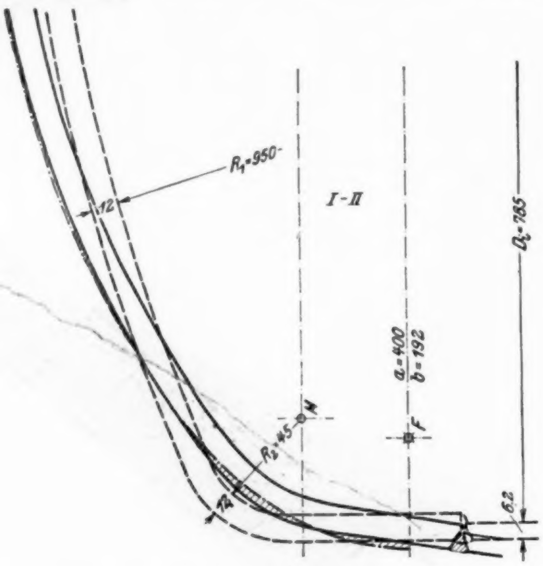


FIG. 3 CROSS-SECTION THROUGH KNUCKLE OF VESSELS I AND II

internal pressure. This should be evident from the fact that the radius of curvature of an ellipse is continuous, whereas in the basket-shaped head it is discontinuous. The best case is obvious, namely, that in which the ellipse becomes a sphere and where the radius of curvature of the knuckle is equal to the radius of the cylinder; the worst case exists when the radius of curvature is very small, that is, where the cylinder has a flat plate for a head. But in spite of all information to the contrary, we still weld basket-shaped heads on to cylinders. This is easy to do by means of oxyacetylene or electric welding. The above facts, however, seal the verdict against this procedure.

Elastic Deformations. In order to discover the conditions as to tension in the heads described above, the strain in the heads due to internal pressure was determined by means of an apparatus designed by Dr. Okhuyzen. This apparatus measures accurately deformations of 0.001. mm. (0.000039 in.) in a simple manner, which accuracy is more than ample for our purpose. The errors in the readings are less than those due to the effect of irregular plate thickness and deviation from the correct shape. Moreover, differences in the elasticity of the plate and stresses due to working and welding are evident. The plate of which the vessel was made was tested and the usual value was obtained and is here used, i.e., E = 2,150,000 kg. per sq. cm. (30,500,000 lb. per sq. in.) and $1{:}m = 0.3 = v$.

For each point the measured strains are taken as a function of the pressure and the starting point rechecked. From the strains the stresses were evaluated, as the former are proportional to the latter; it is only necessary to find a correct proportionality factor.

No matter at what point in the hollow vessel we wish to measure the deformation, if Hooke's law holds, $\sigma = \epsilon E$. First, if $\epsilon = \Delta l{:}1 = 0.001$, then σ = 2150 kg. per sq. cm. (30,500 lb. per sq. in.). Stresses read off from this proportionality are reduced, for the strain in the cylinder is a function of the three principal stresses. However, we have only measured in one direction but we have determined our proportionality factor in the same manner, so we have the correct strain and the reduced stress.

Tensions are shown with plus signs, compressions with negative. The reduced stresses are functions of the strain.

$\sigma_{a\ red}$ = reduced axial tension in shell in plane of projection.
$\sigma_{t\ red}$ = reduced ring and hoop tension in shell (tangential stress) normal to plane of projection.
$\sigma_{m\ red}$ = reduced radial stress in head in plane of projection.
$\sigma_{u\ red}$ = reduced, hoop tension in head perpendicular to plane of projection.

As the plane of projection in this case that plane is taken which contains the axis of revolution of the boiler or the center axis of the shell.

The cross-section of the boiler (Figs. 1 and 2) is developed, using this development as abscissas and thus locating the measuring points. By using the strains or reduced stresses as ordinates the condition of the reduced stress is obtained for any point on the boiler. As shown in Figs. 5 and 6, the ultimate strength of a boiler or other pressure vessel can be determined.

From one point of view this work is limited. It is only possible to measure on the outside of the tank, and therefore only to determine the stresses on the outside. As yet the inside stresses are unknown. The ordinate furthest to the left in each of these tension curves shows the reduced stress in the crown. Theoretically, the hoop tension and the radial tension in the crown are equal at the axis. In spite of this, there are differences in the reduced stresses, $\sigma_{m\ red}$ and $\sigma_{u\ red}$ in the crown.

Irregularities in the shape of the head and in its given thickness are evidently the reason for this. Moreover the stresses never

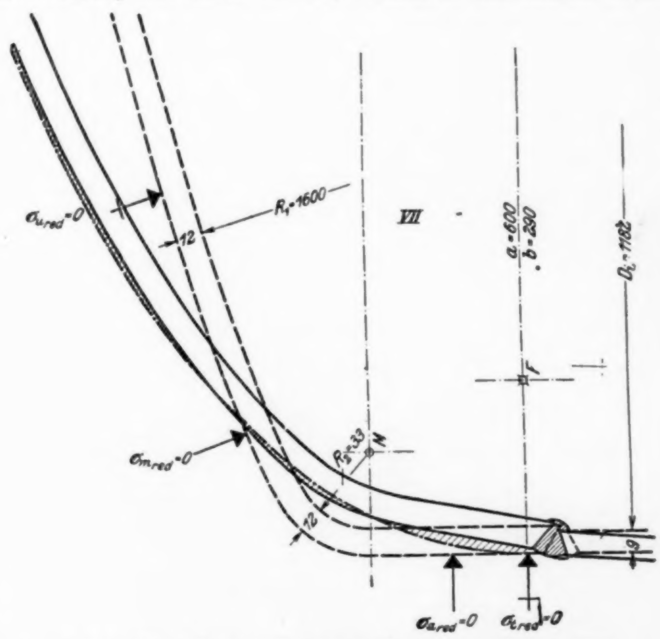


FIG. 4 CROSS-SECTION THROUGH KNUCKLE OF VESSEL VII

distribute themselves exactly as calculated. A glance at the tension curves shows that they are wave-formed and do not change suddenly. In the same general manner, hoop tension reaches a maximum in the knuckle and radial tension at the crown of the head, with another maximum point almost at the same point as the zero value of the hoop tension.

Radial tension and hoop tension change sign near the knuckle, and the compression increases to maximum at the middle of the knuckle. This compression increases as the radius of curvature of the knuckle decreases. In vessel No. 7, R_2 = 3.3 cm. At the knuckle the radial stress and hoop stress are compressions, the radial stress being the lesser of the two. Near the cylindrical shell in the vicinity of the circular weld both stresses again change sign.

The points of change in sign of the stress in Figs. 1, 2, and 4 are designated by black triangles. Both zero values of the radial stress fall between the zero values of hoop stress. This is also true of other hollow vessels.

In the shell, the axial stress (the continuation of the radial stress in the head) falls to half the value of the hoop tension. Before the axial and hoop tensions become constant. oscillations at the beginning of the shell are evidenced.

The circular seams shown in Figs. 5 and 6 by small zigzag lines lie in the vicinity of the maximum values of these axial and tangential stresses, which should be taken into consideration in determining the strength of these seams. They should be reinforced in the same manner as longitudinal seams.

The maximum stress in the elliptical head of vessel No. 8 (Figs. 7 and 8) decreases abruptly (we are dealing with the reduced stresses) as compared with the basket-shaped head in No. 7 which had a very small radius of curvature at the knuckle. The lines

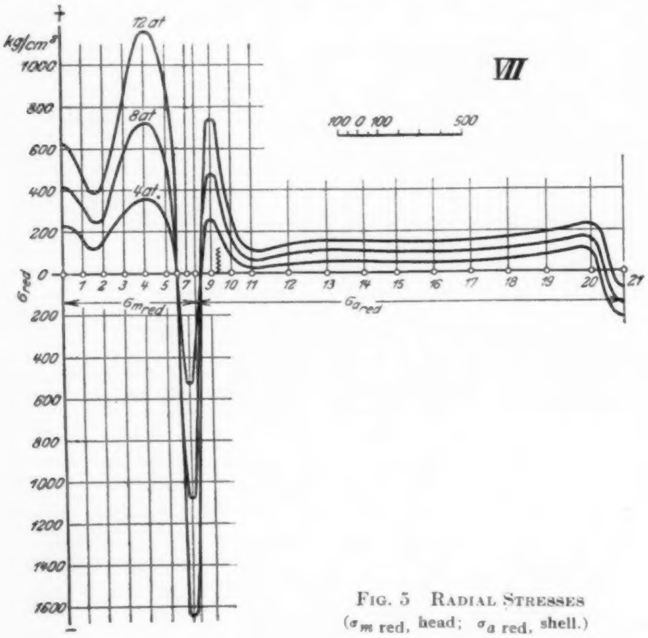


Fig. 5 Radial Stresses
($\sigma_{m\,\text{red}}$, head; $\sigma_{a\,\text{red}}$, shell.)

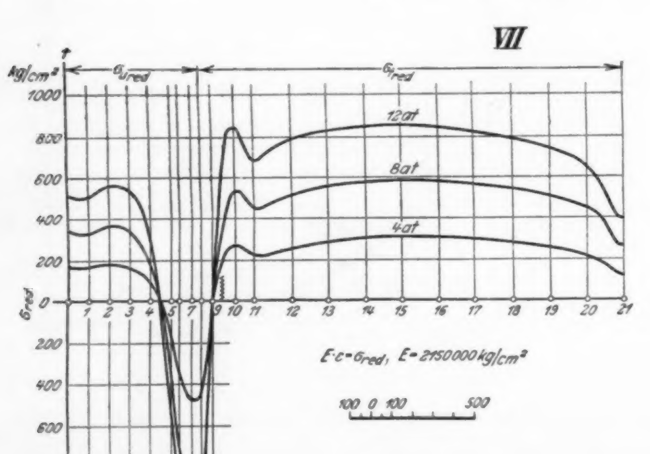


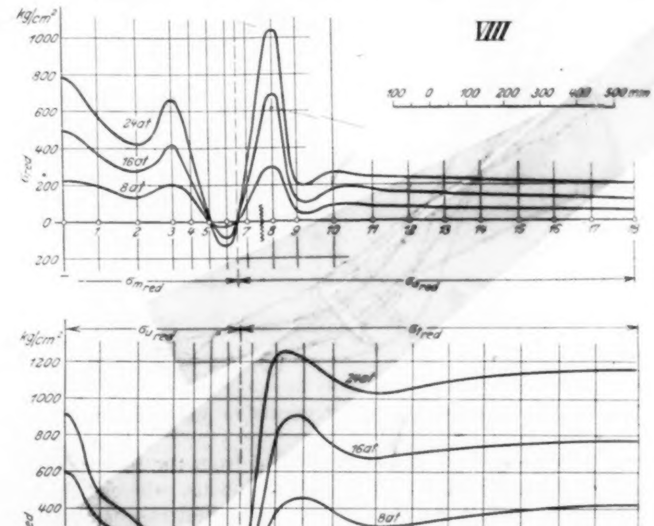
Fig. 6 Hoop Stress
($\sigma_{u\,\text{red}}$, head; $\sigma_{t\,\text{red}}$, shell.)

show a decided increase of stress at the crown of the elliptical head as the head (s = 1.1 to 1 cm. = 0.4 in.) was thinner at the crown than at the cylindrical end (s = 1.2 to 1.3 cm. = 0.5 in.). However, this should not be disturbing, as in the head, where there is lack of metal, the stress is not as high as in the knuckle and around the circular seams. The most economical stress distribution points to a head with elliptical cross-section as the boiler or pressure-vessel head of the future, but the seams must not be weaker than in the usual head.

The change in stress in the shell is much more simple than in the head. The shell is influenced by the head welded on its ends. In the knuckle of the head compressive stresses act on the outside. Thus at the end of the shell there is tension both outside and inside. The effect of the head extends over only a small section of the shell; if the cylinder is long, i.e., if the distance between the circular seams is great, the effect of the head does not reach to the center of the cylinder. In the case of short cylinders the end effects overlap. In the last analysis this is a function of ratio of distance between the circular seams to the radius of the cylindrical shell. In case of vessel No. 8 this ratio is large, so that the effect of the head is negligible before the center of the shell is reached (Figs. 7 and 8).

In the case of vessel No. 7, Figs. 5 and 6, the tangential stresses in the center cross-section of the shell (lower right) decrease and the axial stresses (upper right) change from tension to compression. These peculiarities are due to the butt strap of No. 7, Fig. 1.

It is clear that the compression on the outside must be counteracted by a tension on the inside. Moreover, in the final consideration the tension must be greater, since $\sigma_t = ap/s$ (a = radius of shell, s = wall thickness). In the plotted points bending moments are included. From tests to destruction it has been shown that the plate of the vessel takes its normal form as soon as the



Figs. 7 and 8 Stresses in Vessel with Elliptical Head—Elliptical-Axis Ratio, $a:b = 2:1$
Hoop stress: $\sigma_{u\,\text{red}}$, head; $\sigma_{t\,\text{red}}$, shell. Radial stress: $\sigma_{m\,\text{red}}$, head; $\sigma_{a\,\text{red}}$, shell.

elastic limit is passed by a high internal pressure. The ultimate strength increases and elongation and contraction decrease in the case of three-dimensional stress. The plate acts under three-dimensional stress and has a greater strength in any direction than that of test piece taken from the plate, provided we calculate the strength of the plate as if it were working with unidimensional stress (i.e., the plate strength according to $k_2 = ap/s$).

The Stresses on the Inside of the Head

If we cut by means of two axial cross-sections and two ring sections a short-radius knuckle in a body of infinite extent (Fig. 9), the cutting surfaces are bounded by straight edges and are perpendicular to the median plane of the knuckle. The outer and inner sides of the knuckle are shown by means of full-line and broken-line hatching, respectively.

The section is taken in a place in the knuckle where the reduced ring and hoop stresses are compressions. We know through measurement $\sigma_{m\,\text{red}}$ and $\sigma_{u\,\text{red}}$ on the outside (Fig. 9), and seek the corresponding values of σ_m and σ_u. The stresses perpendicular to the median plane of the head σ_r are neglected after considering their magnitude. Carried through, the calculation shows that the actual stresses as well as the reduced stresses are compressive forces. As we are dealing with stresses in a shell, Fig. 9, we designate outside stresses by e and inside stresses by i.

We can deduce from Figs. 3 and 4 the stress condition of the material on the inside of the knuckle, even though permanent set takes place. The following conclusion as to the permanent set holds in the same way as for the elastic limit, as the knuckle is displaced in the same direction under the elastic limit as above it.

With increased internal pressure the knuckle opens up. From Figs. 3 and 4 we see that in the radial cross-section the outside ring is shoved out (σ_{me} known from measurements and calculation)

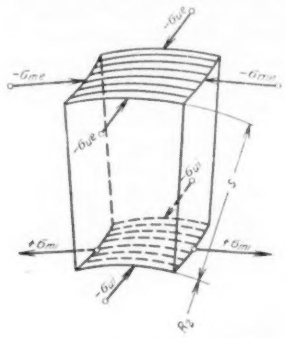


FIG. 9 AN ELEMENT OF THE KNUCKLE

whereas the inner section becomes smaller (σ_{mi} value unknown).

The manner in which the knuckle acts in regard to hoop tensions follows in the same manner in Figs. 3 and 4. On the outside the reduced compression is obtained by measurement. This is also very easy to see, as the knuckle is drawn in by means of the internal pressure. Small circles on the knuckle may become shortened as well as the circles which bound the shell shown in Fig. 9. The middle diameter D_1 becomes shortened to D_1'. The comparative movement $(D_1 - D_1') : D_1 = -\epsilon_{ue}$ below the elastic limit is known, as σ_{ue} is known through measurements and calculations. The circle in question on the inside of the shell becomes shorter in exactly the same way that the outside circle changes from D_2 to D_2'. The shortening $(D_2 - D_2') : D_2 = -\epsilon_{ui} - \epsilon_{ue}$. The fibers in the cross-section are shoved in the same way $(-\sigma_{ui})$ as in the outside ($-\sigma_{ui}$ known), Fig. 9.

$$\sigma_{mi\ red} = \sigma_{mi} - \nu(-\sigma_{ui}) = +(\sigma_{mi} + \nu\sigma_{ui}) \quad [5]$$

$$\sigma_{ui\ red} = \sigma_{ui} - \nu(+\sigma_{mi}) = -(\sigma_{ui} + \nu\sigma_{mi}) \quad [6]$$

In order to evaluate the stresses which the actual elongations require, we must calculate reduced stresses. Two-dimensional calculations, that is, neglecting stresses perpendicular to the middle surface, hold for the knuckle (Equations [5] and [6]), that is, the reduced stresses in the cross-section are increased by the tension in the axial cross-section and the reduced compression in the axial cross-section will be increased by the tension in the radial cross-section. The main thing is to realize that the knuckle on the inside in the radial cross-section is subjected to tension, which, due to the perpendicular compression, is increased so as to be greater than the outside compression. Study has also shown that the knuckles break from the inside and perpendicular to the radius of the head.

The preceding shows that an ellipse is the most practical cross-section for a head. If an ellipse is not used a cross-section should be used which approaches an ellipse as much as possible, and which has a sufficiently great radius of curvature at any point. For safety, boilers and pressure-vessel shells with heads of faulty design should have their working pressures reduced compulsorily, and no obstacles should be placed in the way of such action.

It seems to the author that the stresses should be evaluated according to their effect, namely, the elongations produced, which are easily measured. The magnitude of the elongation is a measure of the danger of failure. We can so arrange the calculations as to determine the elongation consistent with a specific degree of safety. However, if we insist on continuing to calculate with stresses, we must use reduced stresses in our calculations.

APPENDIX NO. 3

Experiments on the Resistance and Change of Shape of Boiler Heads[1]

BY C. BACH

THE experiments dealt with in this paper cover the following:

1 Six boiler heads of elliptical cross-section
2 Six boiler heads of the usual shape
3 Four boiler heads of a shape designed by Engineer Klopper.

They were carried out in the Materials Testing Laboratory of the Technical Institute of Stuttgart and were paid for by the Water-Tube-Boiler Society. The full report of this is given as an appendix to progress report No. 270.

In Figs. 1 and 2, the three head shapes are shown with plate thicknesses of 15 to 25 mm. Head No. 2 is distinguishable by the small radius of the knuckle, which suddenly bends into the radius of the head proper; whereas head No. 1 has a greater knuckle the radius of which gradually changes to the average head radius. Head No. 3 is a mean between No. 1 and No. 2, but more nearly approximates the latter. The diameter of the shells to which the heads were attached was 1300 mm. and the radius of the head in the crown was 1300 mm.

Heads of type 2 were made by the Borsigwerk A.-G. Heads of types 2 and 3 were made by the firm of Phoenix, Abt. Hoerder, Verein.

For those who wish to have a rapid glance at the principal phenomena, the following is given: The maximum stress occurs in the knuckle of the head and is easily evidenced by the

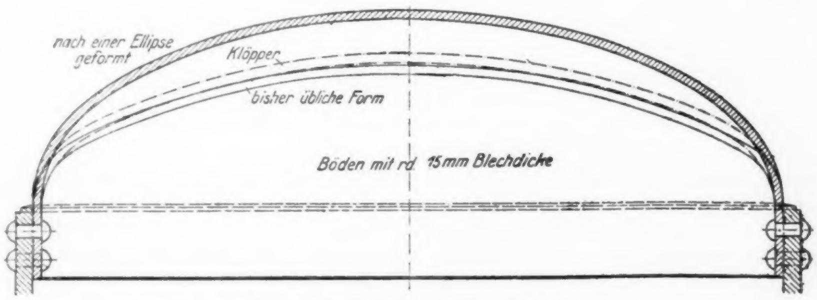


FIG. 1 VARIOUS FORMS OF HEADS FOR ⁵/₈-IN. PLATE
(*Nach einer Ellipse geformt* = elliptical; *bisher übliche form* = usual form.)

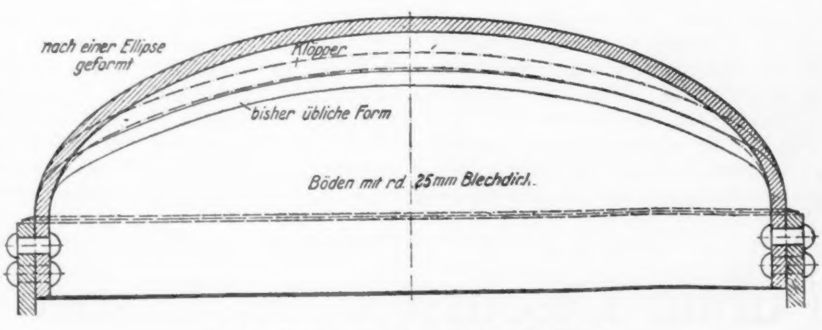


FIG. 2 SAME AS FIG. 1, BUT FOR 1-IN. PLATE

pushing out of the crown. This proves that the elastic limit is reached and exceeded in this section. In the experiments the maximum internal pressure corresponding to the movement of the crown was determined for the different plate thicknesses.

I—ELLIPTICAL HEAD

Plate Thickness	p_{max}
15.6 mm. (0.615 in.)	(42 + 43) : 2=42.5 atmospheres (625 lb. per sq. in.)
19.6 mm. (0.70 in.)	(66 + 62) : 2=64 atmospheres (940 lb. per sq. in.)
24.9 mm. (0.98 in.)	More than 83 atmospheres (1220 lb. per sq. in.)

The pressure could not be increased in the heads with 24.9 mm. plate thickness. As far as could be judged from the displacement of the outside surface of the knuckle, the head would have been pushed out at 90 atmospheres (1320 lb. per sq. in.). Eighty-five

[1] Translated from *Zeitschrift des Vereines deutscher Ingenieure*, vol. 69, no. 12, Mar. 21, 1925, pp. 367–368.

atmospheres (1250 lb. per sq. in.) was taken in order to avoid the use of too favorable a value for the calculations.

II—HEADS OF THE USUAL SHAPE

Plate Thickness	p_{max}.
14.9 mm. (0.58 in.)	(12 + 10) : 2 = 11 atmospheres (162 lb. per sq. in.)
20.3 mm. (0.8 in.)	(24 + 23) : 2 = 23.5 atmospheres (346 lb. per sq. in.)
24.4 mm. (0.96 in.)	(30 + 26) : 2 = 28 atmospheres (412 lb. per sq. in.)

III—KLOPPER HEADS

Plate Thickness	p_{max}.
16.6 mm. (0.65 in.)	(28 + 27) : 2 = 27½ atmospheres (405 lb. per sq. in.)
25.7 mm. (1 in.)	(42 + 45) : 2 = 43½ atmospheres (640 lb. per sq. in.)

The results are graphically presented in Fig. 3. The plate thickness is the abscissa and the pressure at which the plates

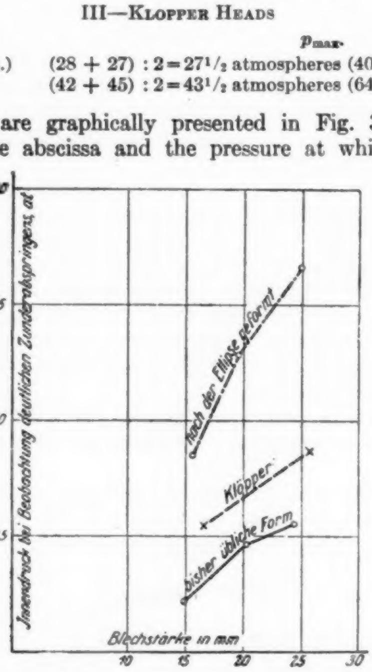


FIG. 3 INTERNAL PRESSURE FOR NOTICEABLE DEFORMATION
(See note to Fig. 1.)

exceed the elastic limit is the ordinate. It is seen that the strength curve of the heads designed according to usual practice lies much below the curve of the elliptical head. This is in accord with previous experiments, and has been shown time and again in the last quarter century for the case of the usual-type heads.

The scaling of the crown of the head (which shows that the elastic limit of the material has been exceeded) is also of importance for the following reasons: The scaling in the case of all heads starts in the knuckle. In the case of heads 2 and 3 with increased pressure it continues to take place mainly in the knuckle (Figs. 53, 54, 62, and 63 in progress report 270), and it only spreads slowly up into the head. This is in contrast to head 1 in which the scaling of the knuckle begins later and distributes itself very rapidly. (Figs. 9, 50, and 58 in progress report 270.) In consideration of the change of shape it should be noted, as shown in Figs. 4 and 5, that the bending in the center of the head in the case of the elliptical head is very much less than in the case of the usual head. As might be expected, the Klopper heads act more like the usual heads.

It should be remembered that this work deals entirely with heads that contain no holes.

In view of the ruling of the German Boiler Committee it is in the interests of safety that the matter of calculations of riveted boiler heads be taken into consideration. The problem may be attacked from the view of theoretical elasticity in a fairly simple manner, as was shown in the International Union of Boiler Safety Societies ten years ago and later reviewed, though even then not sufficiently appreciated. This all deals with the effect of tying up the knuckle with the shell.

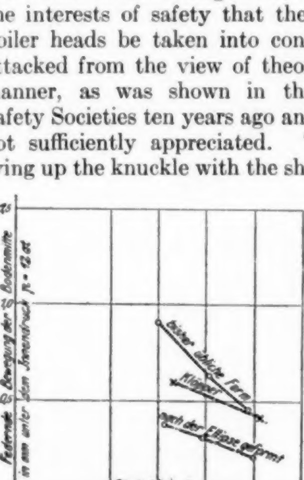


Fig. 4 (Left)

Fig. 5 (Right)

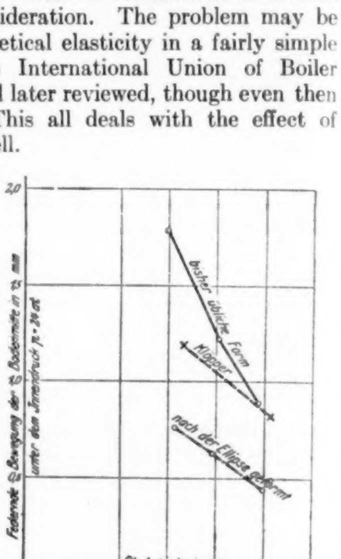


FIG. 4 MOVEMENTS OF CROWN FOR VARYING PLATE THICKNESSES AT 12 ATMOSPHERES PRESSURE (177 LB. PER SQ. IN.)
(See note to Fig. 1.)

FIG. 5 SAME AS FIG. 4 BUT AT 24 ATMOSPHERES (354 LB. PER SQ. IN.)

Because of the riveting of the head to the cylinder shell, we can get much greater or much less tension in the head compared to what we would get if the head were integral with the cylinder. In order to show this without taking into consideration the change of shape of the head under pressure, we can consider the outside diameter of the head somewhat smaller than the inside diameter of the shell; in this case there will be residual stresses because of the fastening on of the head which may be much greater than the stresses obtained under normal pressure. In the latter case the cylinder expands and carries the knuckle with it. In the case of boiler heads which do not remain tight in practice and must be reriveted, the author is of the opinion that the outer diameter of the head was smaller than the inner diameter of the shell.

In a similar manner, deviations from the exact circular form as well as temperature changes, have their effect. In this manner rivet holes can have their shape deformed, from which it may be erroneously deduced that the holes were broached or punched.

Public Discussion on Dished-Head Construction

AN INTERESTING account of a destructive test of a very large fusion-welded tank designed for operation at 300 lb. working pressure appeared in an article entitled, The Design of Dished Heads of Pressure Vessels, by S. W. Miller, which was published in the August, 1926, issue of MECHANICAL ENGINEERING, page 845. In this article promise was made for further data and information concerning the result of investigations then being made on the ruptured head of the vessel, and as a result of the interest that developed the subject was covered in a symposium of four papers which were presented at a joint meeting of the Metropolitan Sections of the A.S.M.E. and the American Welding Society on January 4, 1927, under the following subject heads:

Examination of the Ruptured Head of the Ethylene Tank,[1] by S. W. Miller

Oxyacetylene-Welded Construction of a Large High-Pressure Storage Tank,[2] by H. E. Rockefeller

Stresses in a Large Welded Tank Subjected to Repeated High Test Pressures,[3] by T. W. Greene

The Design of Dished and Flanged Pressure-Vessel Heads,[4] by A. B. Kinzel.

The limitations of this meeting on the evening of January 4 did not permit, however, of sufficient time for complete and satisfactory discussion of the test reports and analyses of the problem of dished-head design, and as a result of urgent requests, the Boiler Code Committee arranged for a public discussion on this general subject on Thursday evening, March 17, 1927. Invitations to the

[1] MECHANICAL ENGINEERING, February, 1927, p. 117.

[2] MECHANICAL ENGINEERING, May, 1927, p. 405.

[3] *Ibid.*, Feb., 1927, p. 124.

[4] *Ibid.*, June, 1927, p. 625.

conference were widely distributed to boiler and tank manufacturers and to all whom it was thought might be interested, and the discussion was based on the reports and conclusions of the four abovementioned papers. An outline of the discussion follows.

The Secretary advised that when the hearing was decided upon by the Committee, he had been instructed to communicate with the chief boiler inspectors of the various states and with the insurance companies to determine whether any of the dished heads that had been built in accordance with the requirements of the A.S.M.E. Boiler Code had failed. One of the results was a list of replies from chief inspectors, all of which make the statement that so far as their records show, none of the heads built according to the Code have failed. He also reported that there is a group of replies from insurance companies in which the same statement prevails.

S. F. Jeter, of the Hartford Steam Boiler Insurance Company, pointed out that heads constructed in accordance with the A.S.M.E. Code had been out only since 1914, and that time was a necessary factor in the production of head failures. He advised that he had a list of head failures that had occurred during the last three years and as far as the information obtained permitted, he had calculated the heads on A.S.M.E. Code formulas to see how the operating pressure corresponded with the head construction used. Some of the heads which failed had been operating at pressures which came within Code requirements. He pointed out that where old heads had failed that were operating at pressures within the Code requirements, they could properly be called failures of Code constructions.

The Secretary continued to report by reading a letter from C. D. Thomas, Chief Boiler Inspector of the State of Oregon, in which the statement was made that in view of the information which had been furnished to the Boiler Inspection Department from outside of the state of Oregon, it was believed that this section of the Code should be revised so that the knuckle of the flange would be equally as strong as any other portion of the head. Further communications were reported from the James G. Heggie and Sons, Joliet, Ill., the A. O. Smith Corporation, Milwaukee, Wis., and F. N. Speller, National Tube Company.

In explanation of a comment in the communication from Mr. Heggie, A. B. Kinzel reported that the condition of the saddle plate had been fully covered in Mr. Miller's report. Mr. Miller had made a very careful examination of the head. The latter was cut up, and strain-gage measurements taken before and after cutting; the thickness of the plate was carefully noted, and this material was also included in the report. No reason could be ascertained as to why failure occurred in any particular place. The thinning out of the saddle plate was small and should have been taken care of by the factor of safety.

Mr. Heggie spoke of the strength of a spherical head, and Mr. Kinzel was not clear as to what was referred to by a spherical head—whether Mr. Heggie meant a true hemisphere or the usual type of large-radius spherical head attached to the cylinder by a knuckle. There was no question that the stresses were less in a true spherical head than in an elliptical head. The reason the latter was taken instead of a true hemispherical head, Mr. Kinzel said, was for economy of manufacture. The stresses in the elliptical head where the minor axis was half the major axis did not exceed the stresses that existed in the shell of the cylindrical vessel with equal plate thickness. In the case of large radius in the spherical head we had not a sphere but an extremely small segment of a sphere joined to the knuckle. The head did not act as a sphere due to the boundary conditions imposed by the knuckle and in order to consider the stresses in the head we must take the spherical section and the knuckle as a unit.

Chairman Low then explained the meeting which was held on January 4, and the apparent need of getting discussion for the benefit of the Boiler Code Committee.

Procedure Followed in Tests

As the author of one of the papers under discussion, S. W. Miller was called upon to explain the procedure followed in the tests conducted. He explained that the tank was built for his company's own use. It was designed carefully and made in accordance with the experience they had had with some tanks of larger diameter but thinner plate. When it was tested, strain-gage measurements were taken, and it was found that around the manhole, particularly at the major axis, there were very serious stresses. These stresses were described in a paper that appeared in the August, 1926, issue of Mechanical Engineering. It might be of interest to note that the stresses at the edge of the manhole at the major axis, under the 900 lb. test pressure, would have been of the order of 870,000 lb. per sq. in. if the steel had stayed within the elastic limit.

On the second test, it was found the head still distorted. The stresses were still great. Those in charge of the test made up their minds that the head was probably not safe, and that they would test it the third time. The first two test pressures were each 1000 lb. per sq. in. In the third test, the head broke at 930 lb. per sq. in. The head was redesigned and a new one made and applied. The test showed that the head stresses were the same as in the rest of the tank. There was no deformation in the welds at any time during the tests, which was very pleasing.

As soon as the head failed, arrangements were made to cut it off back of the weld and send it to the laboratory for examination. Mr. Miller said that he had had the pleasure of doing that part of the investigation which is embodied in the first part of the paper mentioned. The general results were that there was no change in the characteristics of the metal in the head as compared with the mill-test-report figures, in spite of the fact that it had been flanged in a sectional press, and so heated irregularly: heated when welding the manhole flange in, and heated also when the head was welded to the shell.

Mr. Miller called attention to Table 1 of his paper, which gave the results of a number of tests that were made on the pieces taken from all parts of the shell. It was only necessary to glance through the figures to find that there had been no change or alteration in the material. One of the things they wanted to find out was the strength of the weld metal. Fortunately, the head was thick enough so that they could cut out from the weld, proper pieces of the weld metal and test them, although the test pieces had to be smaller than standard. The method of cutting them out was shown by Figs. 6 and 7 of his paper. These showed, first, where the test pieces were taken from the weld metal, and second, that test pieces were taken just as close as possible to the weld in the base metal but not including any of the weld metal.

The results of the pieces taken next to the weld in the plate were given in Table 1. If these were compared with those from the rest of the head, it would be found that there had been no considerable change in the physical properties of the material. In all except one these properties were of the order of the material in the plate. One of the weld-metal test pieces, C-6-A, had a slight defect in it, which accounted for the inability to find the yield point or elongation. The conclusions drawn were that there had been no change in the strength of the plate due to the operations that had been put on it, and there was no reason to be found there for the head failure, and that the weld metal was strong and sound.

Quite a few microscopic examinations of the welds were made to find out if there were defects. In Fig. 8 there was a slight defect at the bottom of the V; this was the worst weld of all those examined. Mr. Miller stated that he wished to impress the fact that this was done in a commercial welding shop, under their company's supervision. One interesting thing was that Figs. 13 and 14 showed the manhole flanges thinned out from what was intended to be $1\frac{1}{2}$-in. plate to about $1\frac{1}{4}$ in., so that it appeared necessary in such heavy manhole flanges that the strength should be figured on the actual dimensions.

In the knuckle and head flange instead of thinning out, the material actually increased in thickness. It was originally $1\frac{1}{4}$-in. plate. This thickened up where it was welded to the shell to about $1\frac{7}{16}$ in. There was one section, No. 1, that included the whole of the head from the weld to the manhole.

As nothing could be found wrong in the material or welding, the only thing left was the design. Mr. Miller said that Mr. Kinzel had worked this out very nicely, and that the test of the redesigned head had confirmed his analysis.

A. B. Kinzel, author of the paper on The Design of Dished and Flanged Pressure-Vessel Heads, then addressed the meeting. He explained that in usual practice the radius of the head was taken as the diameter of the shell and the head considered as a sphere. Strain-gage measurements showed conclusively that the section did

not act as a sphere at all and that we must take the head itself and the knuckle proper as one unit. If this was done one could see at once the extremely high stresses resulting from the common design in certain types of vessels.

Particular attention was called to the stresses about the manhole, which became many times greater than the stresses with no manhole. Mr. Kinzel showed his new design for manhole reinforcement in which the reinforcement was considered as a thick-walled cylinder subjected to external tension, and explained how such a cylinder symmetrically placed with respect to the head plate could be calculated to take care of all the additional stress.

H. LeRoy Whitney, of The M. W. Kellogg Company, stated that he would like to say in corroboration of the work done by Mr. Miller and his associates, that over a period of years in building heavy stills for oil cracking they had discovered some years ago exactly the same points regarding the ordinary design of dished heads. Strain lines appeared in and around both the knuckle and the manhole, particularly at the higher stresses, resulting from test pressures greater than 150 per cent of the working pressure for which the vessel was designed.

As a result of their findings at that time, they had, in the past three or four years, been building all their vessels with a knuckle radius of $^1/_{12}$ of the diameter, 1 in. to the foot, which was a considerably larger radius on the knuckle than the radius prescribed in the Boiler Construction Code. That had helped the situation a great deal, but even with that increase in knuckle radius, they had found that when they went to any high test pressures at all, the head still tended to go into the form of an ellipse, and in every one of the heavy stills they had sent out there had been a permanent deformation in the head after the test. From their point of view there was no question but that the design of the heads ought to be changed.

H. E. Rockefeller, who wrote the paper on Oxyacetylene-Welded Construction of a Large High-Pressure Storage Tank, was next called upon. He stated that they figured the head on the same basis as was commonly used and employed a design fiber stress of 9000 lb., which was less than was allowed in the Boiler Code. They also used a larger knuckle radius than had been the usual practice, and in fact in this particular tank it was larger than 1 in. to the foot of the diameter. He called attention to the fact that in the 5-ft.-diameter tank they used a 6-in.-diameter radius, but that since then the knuckle radius had been further increased so as to approach the true elliptical shape.

In conducting tests they had always used three times the design working pressures, which brought the material up to a stress of 27,000 lb. That was the reason they found out many of the things that had ordinarily passed unnoticed during fabrication, because they never approached that stress in the test, and it was only after they went to twice, and between two and three times the design stress, that they found the large distortions taking place.

With regard to the cover plate and saddle plate, the saddle plate instead of being $1^1/_4$ in. in thickness, as would have ordinarily been used where the metal was taken out of the shell, was $1^1/_2$ in. The cover plate was designed for 600 lb. operating pressure, i.e., 300 lb. above the operating pressure, for they realized that taking the tank up to 900 lb. they considered the cover-plate design which could stand that test pressure. They did not know that a light cover plate might do, but having no precedent they leaned to the safe side and used a heavier one.

As far as the actual welding was concerned, Mr. Rockefeller said, they followed their usual practice of procedure control. Each welder was tested before being allowed to do any welding on the tank. They required test specimens to be submitted. The results were given on the second page of his paper. The welding was carried on very carefully. The welding of the saddle plate was done by preheating. The failure showed the welds were not affected by the failure.

Perry Cassidy then addressed the meeting, stating that he represented a special committee which was appointed by the American Boiler Manufacturers Association to consider this question of the design of dished heads, to attend the meeting, and to do anything possible to coöperate. He called attention to the fact that Mr. Miller and his associates had done an excellent job in the developing the design of pressure vessels, but feared that there might be an implied criticism of the design of riveted drums for power boilers. This was probably not intended, but some people might infer it from the papers and the discussion.

Mr. Cassidy said he wanted to go over briefly some of the points in connection with the test and design of power boilers and learn just what, if anything, was to be drawn from these tests and the papers in regard to changing the design of such boilers. This test had been made on a fusion-welded vessel, the plate being somewhat softer than the plate in power boilers. It had been made that way purposely, because it was more convenient by fusion methods to weld steel of a carbon range under 0.18 or 0.17 than to weld the ordinary firebox or flange steel used for riveted structures.

Mr. Cassidy went on to say that the tensile strength of the steel as ordered was 10 per cent under ordinary firebox steel. This indicated that there might be a reduction of 10 per cent of the strength of the welded drums compared with similar drums made from flange or firebox steel. The steel that was used for American power boilers was, fortunately, due to the foresight of those who made the specifications originally, and the work of the American steel manufacturers, as good for the purpose as any steel that could be obtained. It was certainly better than any steel that was supplied European countries for boiler construction. The Germans, who had published most of the articles on the design and tests of dished heads, used a steel with carbon around 0.08 or 0.09 and with phosphorus and sulphur several points higher than was found in ordinary American practice. That steel was in a range that when cold worked and subsequently heated to a blue temperature, around 300 deg. cent., was put in a brittle and dangerous condition, and due to the high amount of dirt, phosphorus, and sulphur, it was easier to get bad steel in German boilers than in American boilers. The range where that condition stopped was just below the range where the American steel began, and we were not troubled with that difficulty.

The Germans used a factor of safety of 4 in designing their boilers, while in this country a factor of 5 was employed. This probably accounted for the trouble they had had in the failure of heads and the lack of trouble in this country.

Mr. Cassidy called attention to the statement made by Mr. Obert to the effect that there had not been developed a single case of failure of a dished head designed in accordance with the A.S.M.E. Boiler Code from 1914 on. Naturally, the boiler manufacturers did not want to make unsafe boilers. Nor did they want the customers who had been buying these boilers to feel that they had in their power houses boilers that might rupture and cause trouble at any time due to failure of the heads, and unless it was really true, we did not want to do anything that would lead them to believe that their boilers were unsafe.

The Question of Scrapping Dies

He went on to show why it was not desirable to change the design of boiler dished heads unless there was good reason for doing it. In the first place, the present value of the head dies in this country was probably several hundred thousand dollars. That money could not be scrapped without some good reason for doing it. If the shape of the heads was changed, it would be necessary to scrap the dies. Particularly in battery settings, if the depth of the head was increased much beyond what it was now, it would considerably increase the width of the center wall. That increased the cost of the brickwork and the cost of the boiler house due to the extra space necessary. That, Mr. Cassidy said, was one of the greatest difficulties in changing the depth of the head.

Another difficulty was that if anything was done that required thicker plate in drum heads, the thickness of the plate necessary for the higher pressures of the larger diameters would run over 2 in., and some boiler manufacturers hesitated to use plate thicker than 2 in. As it was now, 2-in. plate worked in very nicely at the maximum pressures. Any change in the formula for design of heads would require plate thicknesses over 2 in.

One of the German papers not published in Mr. Miller's series but distributed to the Committee, seemed to indicate that it was not necessary to design a head of an elliptical shape to get the greatest strength for a given depth. Mr. Cassidy felt that the boiler manufacturer would very much rather stick to the basket-shaped head than change to an elliptical head without any difference in the strength. Increasing the knuckle radius would tend

to increase the depth, and it had already been explained why it was undesirable to do that unless there is some good reason for it. It had been known for a long time that the stress in the knuckle around the manhole of manheads was greater than the theoretically designed stress in the heads. As far as that went, it was known that there were a number of other places in the boiler where the actual concentrated stress was considerably greater than the theoretical designed stress.

Following the reasoning that the stress at the edge of a hole in a plate was $2^1/_2$ times the average stress in the plate, the same thing applied to every tube hole in a drum, and there was theoretically a concentrated stress at the edge of the holes equal to $2^1/_2$ times the average stress in the drum plate. Due to the ductility of the plate used, that stress was distributed and tended to be equalized in the remainder of the plate, so that no particular harm came from it and boilers were safely operated with the theoretical stress in the edge of the tube holes considerably greater than the elastic limit of the metal.

Mr. Cassidy said he felt certain that the boiler manufacturers would not oppose any reasonable and necessary change in the design of heads. All that was wanted was that we should keep our feet on the ground and not do anything that was hasty or inadvisable or that had not been proved to be applicable to the particular design of drums for power boilers. The American Boiler Manufacturers Association, he said, were willing to coöperate in any way that they could, and would be glad to make any changes that were necessary, but they wanted to know first that such changes were necessary.

T. W. Greene, author of the paper on Stresses in a Large Welded Tank Subjected to Repeated High Test Pressures, was next called upon. He stated that for several years, in the design and construction of large tanks, the points of local high stress about which they were concerned, were particularly the knuckle and manhole. That they anticipated high stress concentration was shown by the fact that they painted parts of the tank with white cement in order to visualize and photograph the strain lines or Luders lines as they developed on tanks at points of high stress concentration. The photographs (Figs. 9 and 10 of his paper) showed this scaling particularly around the manhole at a pressure slightly in excess of the design pressure.

One of the most illuminating curves of the whole stress investigation of this tank was found in his Fig. 8. These curves showed distribution and intensity of the high stress concentration around the manhole opening. They showed that under the first application of pressure the metal was actually deformed to such an extent that it was overstrained $2^1/_2$ per cent. This was extremely high deformation, and the curves showed that on the second test the metal continued to yield and that failure occurred during the third application of the pressure. Continued yielding under reapplication of pressure was found only at the manhole opening. It might be expected that with this condition failure was inevitable.

The extent and magnitude of the stress concentration around the edge of the opening might be appreciated by comparing the permanent sets measured at this point with those in the shell, which was usually considered the chief concern in tank design. Table 4 of Mr. Greene's paper showed that the permanent set around the manhole was 184 times that of the shell.

Nature, Distribution, and Intensity of Stresses Around Knuckles of Tank Heads

Another point that Mr. Greene wished to bring out was the nature, distribution, and intensity of the stresses around the knuckles of tank heads. The head of a tank having a dish equal to the diameter of the tank and with a comparatively small knuckle radius, deformed under pressure, tending to assume an elliptical shape. The knuckle contracted inward: that was, the diameter of the tank across the knuckle decreased and the center of the head protruded outward. It would be seen that this deformation produces a high bending moment at the knuckle.

In a hoop direction, i.e., around the tank, the metal was compressed and his Table 3 which showed the permanent deformation measured both outside and inside of the knuckle, showed that in this direction the compression was the same throughout the thickness of the metal. It was practically the same inside and out.

In a radial direction, however, the bending resulting from the deformation described, produced stress concentration of a very high magnitude on the inside of the tank. On the outside, the compression from the bending opposed the normal tension so that the stresses were not large on the outside. On the inside, the reverse was true because at this point the tension from the bending was additive to the normal tension stresses, so that their combined effect was extremely large. Table 3 showed that the metal on the inside of the knuckle was deformed in tension about 2 per cent.

It could be appreciated that the deformation around the manhole and knuckle was extremely large when it was realized that the metal was being strained in two or more directions and that the percentage elongation or deformation before failure was decidedly less than when the metal was simply strained in one direction. These results indicated why cracks frequently found in heads of pressure vessels were developed at the inside of the knuckle. High stress concentration in tension on the inside of the knuckle in the radial direction, combined with corrosion and fatigue, it was believed, were the causes of cracks appearing at these points.

G. Donald Spackman, of the Lukens Steel Company, said he was afraid that Mr. Cassidy was wrong in his estimate of dies, as there were several hundred thousand dollars' worth of dies in the manufacturers' hands. When it came to spinning heads, the standard dished head lent itself very readily to manufacture in that small differences of diameter might be made on the same set of formers. On the other hand, in pressing a new form of head a new plug and ring had to be made for every variation in gage, diameter, and corner radius.

The elliptical head, if held strictly to the ratio of minor axis equal to one-half of the major axis, would require a new set of formers for any difference in gage and diameter, provided the heads were ordered to the outside diameter. This would hold true for both spun and pressed heads.

In the case of the standard dished head with a large knuckle radius, the manufacturer always threw up his hands whenever more than 5 in. to 8 in. knuckle radius was specified on light gages. This was due to the rollers being so far away from the former, which greatly increased the chance of buckling. This would likewise be the case in making an elliptical head unless it were pressed. He said that for still work where the gages were between $1^1/_2$ in. and 3 in. there would not be so much trouble from buckling and probably they would be able to handle the work satisfactorily, although the die equipment would run into many thousands of dollars.

Tests on Conical Heads

William R. Kromer, of the H. K. Porter Company, Pittsburgh, Pa., talked on the subject of conical heads which his company manufactures, and displayed several blueprints. Basing their decision on the results of tests made with three different types of heads, in 1909 his company adopted conical heads with manholes for pressure tanks.

Fig. 1, Mr. Kromer said, illustrated a test tank with spherical heads with flanged manholes. This tank had been subjected to internal pressures up to 1300 lb. per sq. in. The tables accompanying the drawing showed the permanent deflection at different locations and pressures. Head No. 1 showed a maximum permanent deflection, $8^1/_8$ in. from the center, of 0.65 in. and head No. 2, of 0.59 in.

Fig. 2 illustrated a test tank with one spherical head on which a heavy cast-steel manhole ring was riveted and a conical head with a flanged manhole. This tank had been subjected to internal pressures up to 1050 lb. per sq. in. The spherical head showed a maximum permanent deflection of 0.12 in. and the conical head one of 0.04 in.

On this conical head they had calculated the maximum fiber stress to be 14,600 lb. per sq. in. at an internal pressure of 900 lb. per sq. in.

Referring next to Fig. 3, Mr. Kromer stated that the following conclusions were reached as the basis of their calculations:

1 That the strength of the successive differential rings constituting the cone, to resist rupture, was that of a hollow sphere of the same thickness having a radius equal to a perpendicular from any point on the surface of the cone to its axis.

2 That as the radius of the corresponding sphere diminished as

the apex of the cone was approached, the stress per unit of section would diminish.

3 That therefore the effect would be the same as though a spherical head were used having an increasing thickness to compensate for the metal removed to form the manhole.

4 That the additional stress in the metal adjoining the manhole would be that due to the desired pressure acting on an area equal to twice that of the triangle *DBF* (if the entire area of the sector were taken it would include area upon which pressure could not exist).

5 That this additional stress due to the removal of the metal to form the manhole would be distributed throughout the metal forming the head between *S* and *E*, reaching a maximum at about *D*, and that as the variable stretching of the metal due to its change of shape within the elastic limit under pressure might be quite accurately represented by a triangle having its apex at *D* and the other two angles at *S* and *E*, it might be assumed as approximately correct that the maximum stress at *D* would be twice the average stress, and that this additional stress would diminish to zero at *S* and *E*.

6 On this basis the maximum fiber stress due to a pressure of 900 lb. per sq. in. in a head of the dimensions given would be:

7 Normal stress at *D* due to the sphere to which that particular differential ring of the cone would correspond.

Radius of sphere = $14^1/_8$ in.

Area of a circle $15\ ^1/_4$ in. radius = 730.62

Area of a circle $14\ ^1/_8$ in. radius = 626.80

Sq. in. section in ring $1\ ^1/_8$ in. wide = 103.82

$$\frac{626.80 \times 900}{103.82} = 5440 \text{ lb. per sq. in.}$$

8 Additional stress due to removal of metal to form manhole.

Twice the area of triangle *DBF*

$7^5/_8$ in. × $11^7/_{16}$ in. = 87.1 sq. in.

87.1 × 900 = 78,390 lb.

Sq. in. of section to resist on both sides of the manhole between *S* and *E* = $(1^1/_8 \times 8) \times 2$ = 18 sq. in.

$$\frac{78{,}390}{18} = 4360 \text{ lb. additional stress}$$

per sq. in. of section if stress were distributed uniformly. But as their assumption was that the maximum stress at *D* would be twice the average, they had 8720 lb.

9 Normal stress at *D*, as referred to in (7)..... 5440 lb. per sq. in.
Additional stress at *D*, referred to in (8)..... 8720 lb. per sq. in.
Maximum fiber stress at *D*............... 14160 lb. per sq. in.

Shape of Head Not a Safety but an Economic Question

Wm. H. Boehm, representing the boiler insurance department of the Fidelity and Casualty Company, stated that Mr. Miller had shown that for a given kind and thickness of material the tests made indicate that a German head of elliptical shape was stronger than an American head of basket shape. Mr. Cassidy had shown that a basket-shaped head designed with knuckles of relatively large radius might be as strong as an elliptically shaped head. Mr. Walker had shown that cone-shaped heads had proved satisfactory in some instances. Statistics seemed to indicate that no American head of basket shape designed in accordance with the A.S.M.E. code had failed for lack of strength. This question of shape, therefore, was not a safety question, but an economic one, and Mr. Boehm was not sure that it would be in the interest either of the user or of the manufacturer to scrap thousands of dollars worth of dies merely to change from one shape to another. Since American heads as at present designed were not unsafe, no Code rule should be adopted that would prohibit any manufacturer from

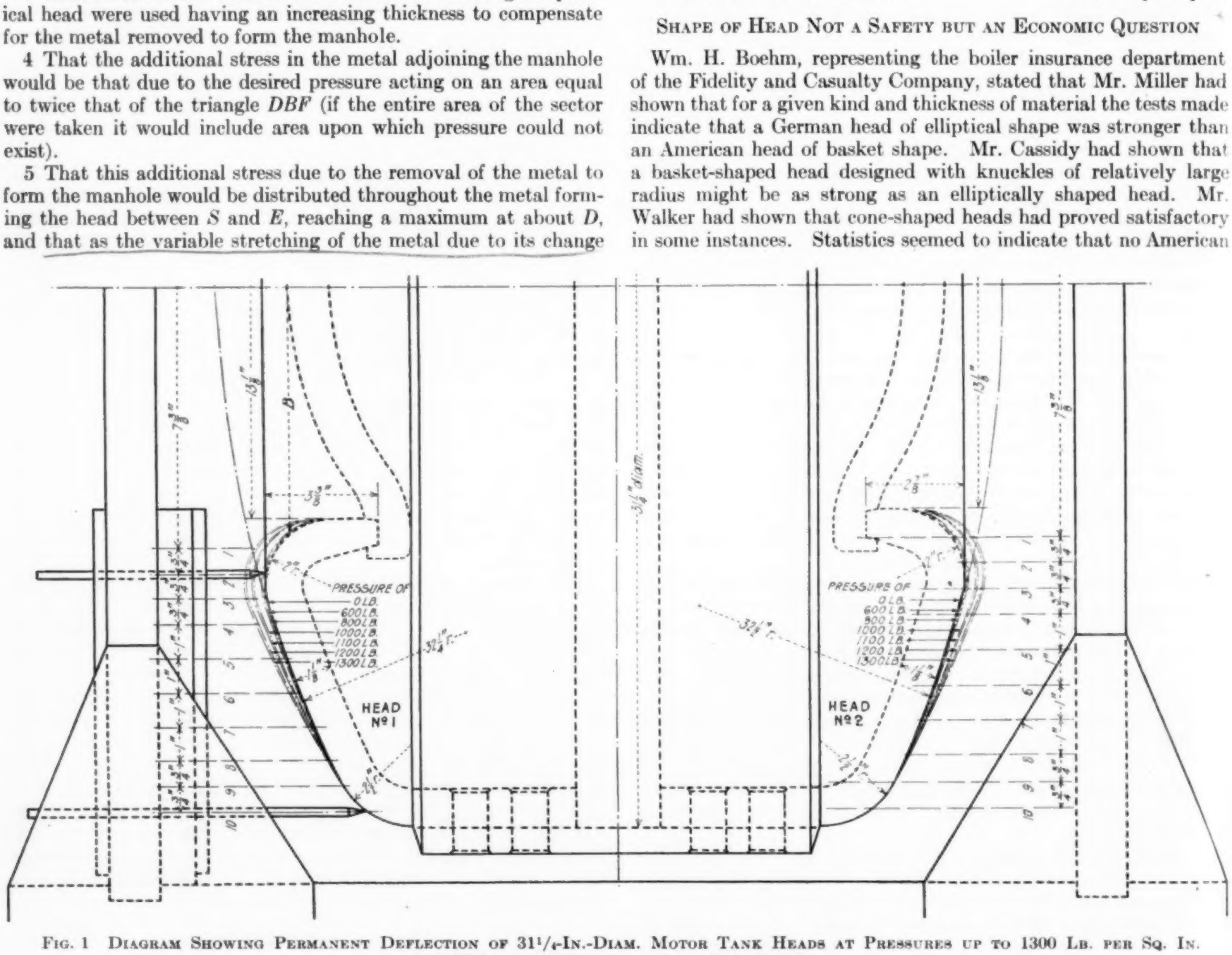


Fig. 1 Diagram Showing Permanent Deflection of $31^1/_4$-In.-Diam. Motor Tank Heads at Pressures up to 1300 Lb. per Sq. In.
(For deflections and dimensions of *B*, see Tables 1 and 2.)

TABLE 1 DEFLECTIONS OF HEADS IN INCHES

Rod No.	Pressure, lb. per sq. in. 0	600	800	1000	1100	1200	1300
Head No. 1							
1	...	0.04	0.07	0.36	0.42	0.60	0.65
2	...	0.04	0.07	0.36	0.42	0.60	0.65
3	...	0.04	0.07	0.33	0.375	0.546	0.61
4	...	0.02	0.06	0.29	...	0.47	0.535
5	...	0.02	0.045	0.21	0.23	0.35	0.405
6	...	0.01	0.03	0.14	0.16	0.225	0.265
7	...	...	0.02	0.08	0.09	0.135	0.15
8	...	...	0.015	...	0.03	...	0.06
9	...	...	...	...	...	...	0.02
Head No. 2							
1	...	0.032	0.078	0.355	0.40	0.55	0.59
2	...	0.031	0.077	0.34	0.39	0.54	0.59
3	...	0.031	0.062	0.315	0.36	0.48	0.535
4	...	0.031	0.055	0.25	0.28	0.375	0.42
5	...	0.02	0.04	0.15	0.18	0.25	0.28
6	...	0.01	0.02	0.10	0.11	0.15	0.17
7	...	0.005	0.01	0.03	...	...	0.06
8	...	...	...	...	...	...	0.01

TABLE 2 DIMENSION "B" OF HEADS NOS. 1 AND 2

Pressure, lb. per sq. in.	0	1000	1300
Head No. 1	$13^3/_{32}$ in.	$14^3/_{16}$ in.	$14^9/_{32}$ in.
Head No. 2	$13^{27}/_{32}$ in.	$13^{31}/_{32}$ in.	$14^3/_{32}$ in.

making an elliptically shaped head or any other shape desired. In other words, the Code as respected the shape of these heads should remain as it now stood. The manufacturer should be permitted to use any design that provided the required factor of safety.

T. H. Walker, representative of The Baldwin Locomotive Works, said that while he had nothing but memory to quote from he would endeavor to tell of a case in the practice of his company. He said they had built a tank about 34 in. in diameter, $^{13}/_{16}$ in. thick. One head had a manhole. The radius of the head was equal to the diameter of the tank, or about 34 in. The other head was made the same, excepting the manhole opening. Both heads were $^{13}/_{16}$ in. thick. The manhole was reinforced by a flange $4^3/_8$ in. deep. The tank was to carry an air pressure of 900 lb. per sq. in.

The shop had been instructed not to turn the outside of the head flanges, but to use a die whose diameter outside was true, and which was of the same diameter as the tank. In place of this, the shop found a die which was about $^3/_8$ in. in diameter greater than the diameter of the tank. They put the manhole head through this and then turned the flange down to $^5/_8$ in. thick where it went into the tank. The plain head was made to fit the tank without turning. Both heads were riveted in place before the disregard of orders was discovered. They therefore concluded to investigate its behavior under loads above the pressure it was intended to carry, and so they started to run the pressure up. When there was about 1100 lb. per sq. in. on the tank, the manhole flange started to turn inside out, protruding the head in the center around the manhole and reducing the depth of the reinforcing flange, so that one could take the nut of the manhole-cover bolt and screw it up with the finger several turns. Just about the time that happened, the tension on the plate bent the flange that was turned down thin, so that the whole flange pulled away from the shell, in effect increasing the knuckle radius. Before the test, templets were fitted against the heads. During the test, the plain head apparently did move about $^1/_{16}$ in. but after the pressure was released it went back so that no change in shape could be seen.

On the manhole end, the head changed its shape so that it became a tangent from the manhole to the knuckle radius. He said he put his rule on it and it was practically a straight line. The defective head was replaced by one with radius of dish about 82 per cent of the tank diameter and of full thickness of flange. It then stood the pressure test without change of shape and was put into service, and so far as was known today it was still in service.

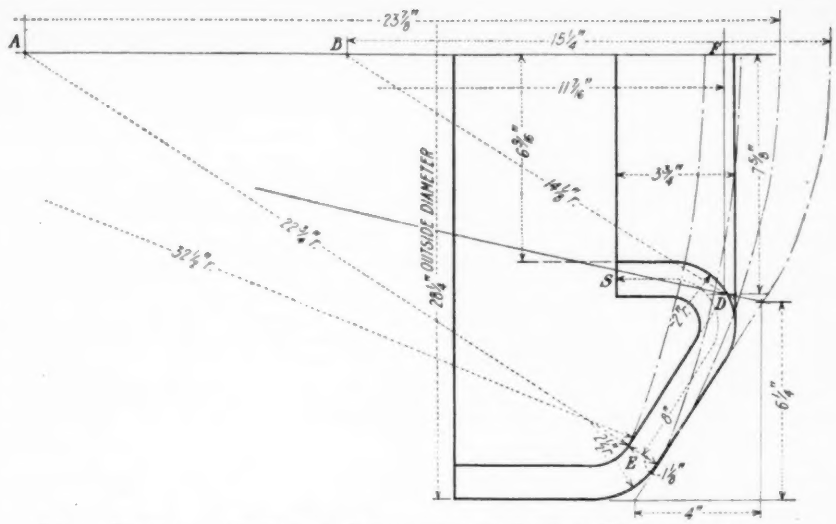


FIG. 3 DIAGRAM ILLUSTRATING EXPLANATION OF CONICAL HEAD WITH MANHOLE

Some one might ask why the heads on this tank were not made heavier. The reason was because the weight was limited by the capacity of the cable by which it was to be lowered into the mine which it was to serve. Mr. Walker said it seemed to him that the shape taken by the head with the manhole when under pressure, namely, a frustum of a cone, was the correct shape for that type of head and that this should be made with a knuckle radius oftentimes greater than four times the thickness of the head.

He stated that he believed the reverse of this was true for dished heads with pressure on the convex side. Here the knuckle radius should be as sharp as possible, and if a heavy reinforcing band was placed outside of the body of the tank opposite the head flange, the head rivets fastening all together, it would prevent the head from reversing.

FURTHER GERMAN EXPERIMENTS ON HEADS

P. W. Swain, of the editorial staff of *Power*, read abstracts of two German papers, one by Dr. Ing. Wilhelm Otte, and the other by Messrs. Erich Siebel and Friedrich Korber, which had been translated by the editorial staff of MECHANICAL ENGINEERING.

The article by Dr. Ing. Wilhelm Otte in the November 30, 1925, issue of the *Zeitschrift des Bayerischen Revisions-Vereins*, sustained the thesis that the elliptical profile of dished heads was not the best. Dr. Otte suggested that the head profile should be that of a "basket;" that was, the customary approximation of a half-ellipse, consisting of a large-radius crown arc connected to the cylindrical portion of the tank by knuckle arcs of smaller radius. With the same ratio of long to short axis, he pointed out that the basket shape gave a larger knuckle radius and a smaller crown radius. The following paragraphs constitute a brief abstract of Dr. Otte's paper:

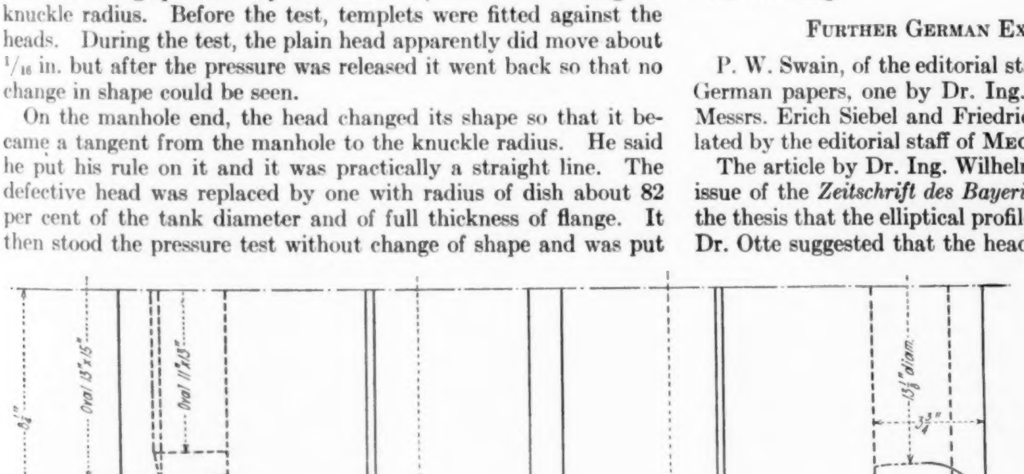


FIG. 2 DIAGRAM SHOWING PERMANENT DEFLECTION OF $38^1/_4$-IN.-DIAM. MOTOR TANK HEADS AT PRESSURES UP TO 1050 LB. PER SQ. IN.

This discussion is concerned only with unstayed dished heads having the pressure on the concave side. It has been the custom to design such heads on the assumption that the head is stressed as part of a hollow sphere having the same radius as the body of the head. This assumption is based on the further assumption that the flange radius is large enough to insure a gradual transition from the crown to the cylindrical part at the circumference of the head.

No rational basis has been developed for determining the proper radii for crown and knuckle. As a

result of numerous accidents, indicating that existing standards were unsafe, Klöpper's proposals, prescribing that the radius of curvature should not be greater than the inside shell diameter, and that the knuckle radius should be, as a rule, at least one-tenth of this, were used as the basis of preliminary official regulations.

This ruling shows progress, but the question is by no means closed, and it would be regrettable if the shape of heads were to be fixed legally before the definite clearing up of these conditions. Recent experiments by Bach have proved that heads of elliptical section and with a ratio of two to one for the main axes (depth of head one-quarter the diameter) are about four times as resistant as the heads hitherto used and about twice as resistant as the Klöpper heads.

Attention, however, must be drawn to the fact that the elliptical form is not, as such, the cause of increased strength, as might be concluded from Bach's publication. It is due entirely to the more favorable dimensions of the crown radius and the knuckle radius, for the chosen ratio of the axes. Generally, at a given ratio of depth of head h and diameter d, that form is the most favorable in which the crown radius r_1 is a minimum and the flange radius r_2 equals $0.125d$ for an ellipse.

For the same case, where h is equal to one-quarter of d, the best proportions of a basket profile are as follows: r_1 equals $0.9048d$ and r_2 equals $0.1726d$.

Here the crown radius is 9.52 per cent smaller than in the case of the corresponding ellipse. On the other hand, the knuckle is 38.1 per cent larger. This larger knuckle radius favors the basket curve over the ellipse.

The following is a brief summary of a report made by Erich Siebel and Friedrich Körber on experiments made at the Kaiser Wilhelm Institute fur Eisenforschung zu Düsseldorf to secure evidence as to the effect of manholes on the behavior of boiler heads when under internal pressure. The report also discussed results of previous experiments on the strength of dished heads without manholes.

The investigation made by the writers may be outlined as follows: (*a*) measuring the elastic deformation perpendicular to the surface and determining inception of permanent deformation as well as the permissible test pressure for measurements under *b* and *c*; (*b*) measuring the changes in curvature; (*c*) measuring the dilatations of the surface; (*d*) measuring the permanent deformations and observing flow phenomena; (*e*) measuring elastic deformations after previous strong permanent deformation.

[The discussion of the methods of testing used and the results obtained as given in the original paper, is rather technical. Moreover, the conclusions reached are highly complex, involving a great number of factors. It is therefore impracticable to do more here than indicate in a very general way the main trend of the conclusions.—Editor.]

The conclusions reached from the study may be summarized as follows:

1 Strains in heads without manholes are due in the first place to stress maxima appearing in the direction of the meridian at the inside of the knuckle and at the outside of the main curvature. These are caused by strong bending loads, due to curvature changes in the meridian curve. Heads of elliptical shape, and those of basket profile closely approaching the elliptical form, show the lowest stress values for a given depth of head.

2 In manhole heads, extraordinarily high stresses occur at the manhole edge, far surpassing the maximum stresses in corresponding heads without manholes. Heads developed with a protruding manhole edge show a considerable reduction in edge stresses and deformations.

3 The deformation of the heads is in agreement with the stresses described. At the knuckle the elastic line always has a strong negative curvature, corresponding to the high bending moment existing there. At the edge of the manhole both the stresses and the deformation are greatest.

The experiments show that the stresses around manholes far exceed those imagined by designers, so that in many cases the material is locally stressed up to the flow point when the drums are subjected to ordinary working pressures. This indicates the extreme importance of further study along this line and of new standards for the design of dished heads, particularly heads containing manholes. The bad effect of manholes is exaggerated when they are placed off center or when they are of elliptical rather than of circular section.

A.S.M.E. Code Rules for Solid Heads Reasonably Safe for Boiler Practice

Mr. Jeter went on to express the hope that the Committee would not be stampeded into making changes. He said that there were a number of practical considerations that should be taken into account besides the theoretical one regarding the strength of the heads. A sphere was the strongest form in which a container could be built, and he believed the first boilers were practically spherical. But this design was soon abandoned and was dropped down to the second form of strength, which was a cylinder. A manufacturer was not now forced by the Code to use a cylindrical design. He could build a rectangular box header, or other form, so long as he built it strong enough. The question as to whether some elliptical form or approximate elliptical form should be adopted was only a question of whether we were trying to distribute the material in the most economical manner, and in gaining that economy in material we might lose in a great many other ways.

There was one saving point in the matter of head failures, Mr. Jeter said, and that was that the failure of a head was a matter of slow progress, and the failure generally occurred at a point where reasonable inspection detected the tendency to fail before a point was reached where actual disruption of the vessel occurred. He said his company's inspectors discovered many heads that were failing. In practice it was rare that a head went to the point of actual failure, and when they did, every case that had come to his attention had been on account of neglect or improper inspection.

Mr. Jeter went on to say that it seemed to have been implied in the discussion of figuring the stresses in heads, that it was assumed by the Code Committee that the heads could be figured as spheres, but he thought the rules showed this was not so. If heads were made of the most economical form as far as material was concerned, it might be found that the heads were so thin that when used in places where they would be subject to corrosion there would not be enough material in them to make them lasting. Many cases that would ultimately have been head failures were due to the fact that the head corroded from the outside until it was much thinner than it was originally designed. The thinning produced a condition so that the stresses were increased at the points where cracks occurred. Of course, as all knew who worked on the Code rules, they were really too liberal when applied to large-size heads. The Code rules were very much safer in sizes, say, up to 42 in. than they were for 60 in. and higher diameters, and consequently heavier material.

Mr. Jeter said he felt that the Code rules for the solid heads were what might be called reasonably safe for boiler practice. For manhole heads, he felt that in some cases the rules produced a design that carried too high stresses. He then reported on 15 cases of head failures with data compiled by his company, these having occurred between 1923 and 1926.

With regard to failures of A.S.M.E. Code heads, Mr. Jeter said he believed that there would ultimately be some failures of such heads, at least cracking, just as had been experienced with heads of the same design that were built before the Code was in existence. Of course, none of the heads which had been reported on, with possibly one exception, were in boilers built under A.S.M.E. Code rules.

He again repeated the hope that the Committee would not prevent a manufacturer building a basket-shaped head if he wanted to, provided he made it strong enough. It seemed that the only place where the Code rules for boilers were a little weak was in regard to the heads with manholes. However, so far as his experience went, there was not a predominance of failures of manhole heads. He said that he understood from the discussion, that a manhole head was supposed to be weak through the manhole, and that he could not remember of knowing of an explosion where a dished head had failed through the manhole. Failure always occurred in the inside of the flange. If there was a sharp turn at that point, a crack would always occur there unless the head was enormously thick.

G. F. Nordenholt, of the American Foundry Company, New York, N. Y., expressed the opinion that the Committee should first consider whether it was desirable to change the A.S.M.E. Code for Unfired Pressure Vessels, and should investigate the present formulas as set forth in the Code to determine whether they were logical for all cases to which they might be applied.

He was of the opinion that if the present formulas were confined in their application to boiler drums of diameters commonly used, the relations set up would hold out pretty well.

He raised the question as to what the effect would be if these same rules and formulas were to be applied to tanks 60, 70, and 80 in. in diameter. He also brought up the point that there was nothing specified in the Code regarding the relation between the radius of curvature of a spherical head and the diameter of the tank. It was probably assumed that the radius of the spherical head would be made equal to the diameter of the tank, which was common practice, but not followed out in all cases.

As regards the knuckle radius, he brought out that the Code for Unfired Pressure Vessels specified that for plate thicknesses up to $^1/_2$ in. the knuckle radius should not be less than three times the plate thickness, and for thicker plates, not less than 3 per cent of the radius of dish, and in no case less than $1^1/_2$ in. Thus, with a radius dish of 120 in., a plate $^1/_2$ in. thick would require a corner

radius of $1^1/_2$ in., while for a plate $^9/_{16}$ in. thick this corner radius would have to be 3 per cent of 120 in., or 3.6 in. This did not seem logical.

In some of the German technical papers it had been brought out there was little to be gained between the true elliptical head and a basket-shaped head, provided the heel radius of the basket-shape head was of the proper proportion.

In a paper by E. Siebel in *Stahl und Eisen*, Sept. 2, 1926, a relationship was given regarding the total depth of the head, the heel radius, and the spherical curvature, for basket-shaped heads. The total depth of the head in this case was taken as the axial distance between the extreme inside point of the dished head and the point where the dished head became tangent to the cylindrical portion of the tank. Mr. Siebel stated that in order for a basket-shaped head to approach in its stresses a true elliptical shape head having a major and minor axis in the ratio of two to one, it was necessary that the heel or knuckle radius of the basket-shaped head be equal to or greater than twice the square of the depth divided by the inside diameter of the tank, and the radius of curvature of the spherical portion of the head be equal to or less than the square of the diameter divided by four times the depth.

In one of his papers Mr. Siebel presented a curve based upon numerous experiments, showing that there was little to be gained in going to the elliptically shaped head as compared with a properly proportioned basket-shaped or spherical head.

Mr. Nordenholt expressed the opinion that if the elliptical head were adopted it would mean a tremendous scrapping of dies, as there would be a different shape of head for every different diameter of tank. It appeared that if the German tests had any value it would not be a practical thing to demand a true elliptically shaped head in boilers. On the other hand, if the present design of spherical heads was changed only to the extent of calling for a specific range for the heel radius and a specific range for the spherical radius, many of the dies now in use might possibly be altered to meet the new design. This would be possible in the cases where the wall thickness of the present die was great enough to permit its being turned down to the new shape. In order that this might be done it was of course necessary that the new shape required should not diverge greatly from the present one.

Revision of Code Rules Desirable in Some Cases

He said he felt that whatever the A.S.M.E. Boiler Code Committee might do in the matter of revising the dished heads, should be done very carefully. Undue importance should not be attributed to the elliptically shaped heads. If spherical heads were retained, more attention should be given to the heel radius and definite proportions, or at least a range of definite proportions between the radius of curvature of the spherical head and the diameter of the tank and the corner or heel radius should be established.

He further expressed the opinion that some revisions should be made in the present A.S.M.E. Code as in some cases the formulas were not logical when applied generally. He used for an illustration the present formula upon which the thickness of the dished head was based. This formula had in its numerator a factor which was the radius of curvature of the dished head. If an attempt were made to apply this formula generally, it would result in excessively thick heads for dished heads having a large radius of spherical curvature. As a matter of fact, if the formula were applied to a flat head whose radius of curvature was infinity, the thickness of the head sheet thus calculated would be infinity.

H. J. E. Banck, of the Springfield Boiler Co., Springfield, Ill., said that that company was experimenting at the present time with cast-steel heads. They had not completed their tests, but he showed a blueprint which gave the shape of the head for 750 lb. steam pressure—see Fig. 4. In this particular case, the inside shape of the head was elliptical. The change in metal thicknesses between the greater thicknesses around the manhole opening and thinner skirt flange was gradual and of a shape to comply with the requirements of the foundry. Another feature of interest was that the feed and water-column connections were cast integral as a part of the head, thus eliminating riveted nozzles.

He said that as soon as they had completed their tests, they would be glad to submit the results to the Boiler Code Committee. He also stated that if the Union Carbide and Carbon Research Laboratories should desire one of these heads for test purposes, his company would be glad to supply it.

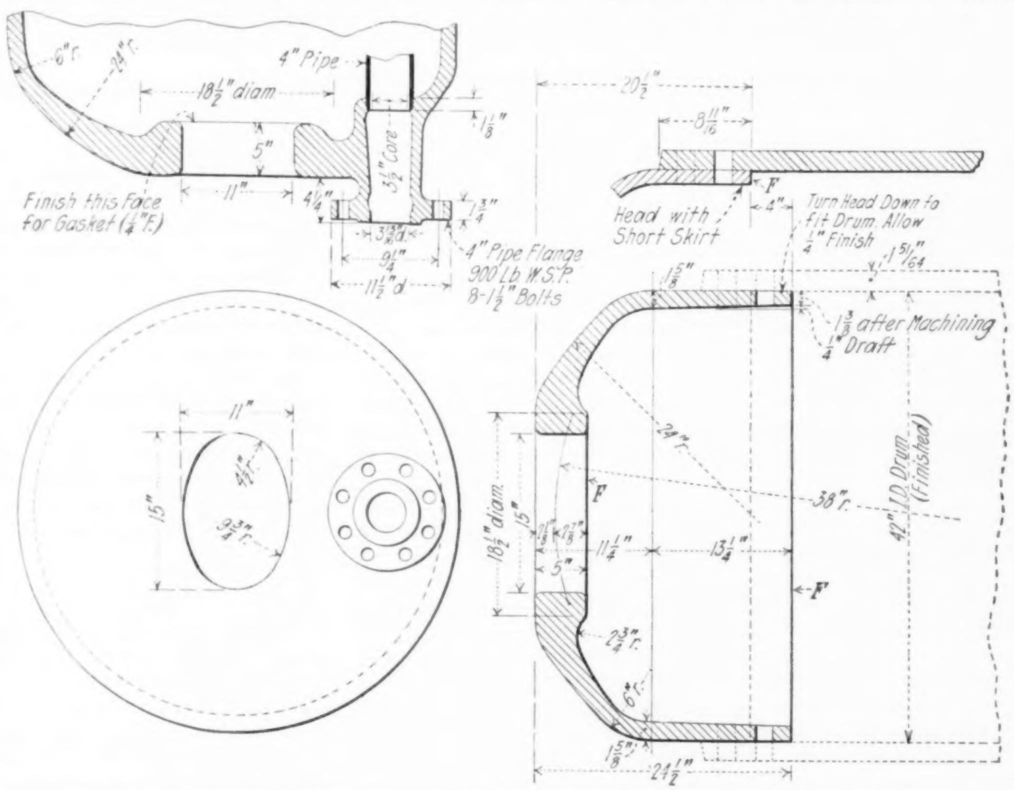


Fig. 4 Proposed Cast-Steel Head for 42-In.-Diam. Steam Drum for 750 Lb. Pressure

S. Mensonides, of Farrar and Trefts, Inc., said that he was thoroughly in accord with everything Mr. Cassidy had stated in behalf of the American Boiler Manufacturers Association, but there was one point he would like to call the attention of the Boiler Code Committee to, if they should feel compelled to change the rules. He said that what held good for high-pressure boilers with small diameters, did not necessarily hold good for low-pressure tanks with large diameters. When one considered a large-diameter tank of low pressure, the knuckle and dishing radius proportion of 1 to 10, as suggested in some of the papers, seemed to be absurd. For a tank of 10 ft. diameter, pressure up to 75 lb., this procedure would give a knuckle radius of 12 in., which from a manufacturer's standpoint would be very impracticable. That was one point on which he would ask the Committee that some exception be made for low-pressure tanks in formulating any possible new set of rules.

Mr. Jeter said he felt the question of cost of dies would be settled if rules could be made for calculating the strength of all the different-shaped heads that might be required, so that the manufacturer could figure the strength of a head of a given form and then the question of cost of dies for making such a head would take care of itself. There must be a great saving of material or some other advantage gained to use a new form of head. It would be wrong

to day that a manufacturer had to make a head of a given form. He should be given the necessary formulas and allowed to use any shape that best suited his purpose so long as he made it strong enough to be safe.

Mr. Miller said that he would like to point out one thing in connection with boiler and pressure-vessel rules. He felt that he was not mistaken when he stated that the ordinary rules for the thickness of heads were devised a long time ago, and that they were probably not far from right for thicknesses then in use, but the increase in thickness of boiler-drum heads and other similar structures had been very great and very rapid recently.

The formulas provided in a head without a manhole for an increase in the thickness of the head as calculated, of $^1/_8$ in. In a $^1/_2$-in. head that $^1/_8$ in. was 25 per cent increase. On a head $1^1/_8$ in. thick it was an increase of only about 11 per cent. If the thickness of the head was increased by a factor instead of by a definite amount, the head would be a good deal stronger. In a manhole head the thickness was still further increased by $^1/_8$ in., so that if a plain head were $^5/_8$ in. thick, a manhole head would be $^3/_4$ in. thick.

In the tests as made, the tanks for his company's use were not only tested at the standard boiler test pressure of $1^1/_2$ times the working pressure, but at 3 times the working pressure, which might develop weaknesses that would not appear under the ordinary $1^1/_2$-times test. If proper design were followed, no greater thickness of head would be needed than of shell.

There could be no objection to making the outline of a head with circular arcs closely approximating the true ellipse. Probably such a head would, under test, take the elliptical shape.

The Influence of Elasticity on Gear-Tooth Loads

Progress Report No. 4 of the A.S.M.E. Special Research Committee on Strength of Gear Teeth[1]

It is always difficult to decide when and what to report on unfinished research. Since our last report, several series of tests have been made. The study of these new data together with further study of the data obtained on the first series of tests has made evident the need for a detailed analysis of all phases of gear-tooth action under load before these test results can be properly interpreted. All of these tests, however, indicate very definitely the important part played by the elasticity of the materials.

It was originally assumed that the test loads represented those that would just hold the teeth of the gears in contact, and as under these conditions the load between the teeth would be reduced to zero for an instant, that the maximum load on the gear teeth would be very close to double the test loads. The original analysis of the test results was made on the basis of this assumption.

It has become very evident, however, that the maximum loads on the teeth have been greater than double the test loads, because on several tests the surfaces of the teeth began to show signs of distress under much smaller loads than should be expected. On further consideration it is evident that the test loads are those required to maintain the electrical circuit through the mesh of the gear teeth. Under static conditions this circuit is not broken until the teeth have been separated a distance very close to 0.002 in.

The influence of errors on the gear-tooth profiles is to cause accelerations and decelerations when running. When the acceleration is sufficient to cause the teeth to leave contact, they will come together again with an impact, the force of which may be many times as great as the applied load.

Thus it is evident that under these conditions it is necessary to make a careful analysis of acceleration loads, impact loads, and the influence of the deformation of the tooth profiles under load in order to interpret the results of the tests. It appears at present that the test loads are a measure of the acceleration loads.

The Committee therefore intends to give in the next few progress reports basic analyses of the influence of elasticity on (1) perfect gears; (2) acceleration loads on imperfect gears; and (3) impact loads. These reports will be issued as rapidly as the material can be developed. This present report gives the first of these studies.

Mr. Carl G. Barth has been giving freely of his time to assist the Committee in this work, and has not only assisted in making some of the tests but has also made considerable progress in the analysis of impact loads, probably the most difficult problem we have to solve. As a matter of fact, it has been largely through his suggestions and assistance that logical methods of analysis of all of these load problems have been made evident.

One series of tests has been made to determine limiting loads for wear for various materials. Since these tests were carried out, however, it has become evident that further tests are necessary to complete this study. Such further wear tests are also necessary to check the accuracy of the various load analyses which are now being made. These tests will be made at the earliest opportunity.

At the present time two other series of tests are being made: one to determine the influence of masses on the tooth loads, and the other to determine the friction losses.

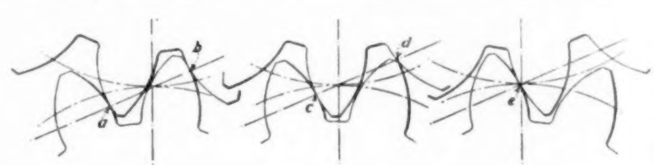


Fig. 1 Mesh of a Pair of Gears in Different Positions

I—Perfect Gears

Involute gears made of rigid materials, rigidly mounted, and with perfect tooth profiles perfectly spaced would transmit power smoothly and continuously at any speed without any variation in the total pressure between the teeth, regardless of whether a single pair or two pairs of mating teeth were in contact.

With elastic materials, however, there would be a certain amount of deformation caused by the load. This deformation would be due partly to the bending of the teeth and partly to the compression of the material. Furthermore this distortion or deformation would not be constant at all phases of the tooth mesh.

This variation in the amount of deformation is due to several causes. First, the point of contact travels over the active profiles of the mating teeth, thus applying the load at different distances from the base of the teeth and causing different amounts of bending. Second, at some points two pairs of mating teeth would be dividing the load, while at other points but a single pair of mating teeth would be in action so that the whole load would be concentrated on them. This condition is illustrated in Fig. 1, which shows the mesh of a pair of gears in three different positions.

The foregoing conditions would result in a variation in the amount of deformation as the contact traveled over the active profiles of the teeth. Therefore, even with perfectly formed and spaced gear

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Wilfred Lewis, *Chairman*, President, Tabor Manufacturing Company, 6225 Tacony Street, Philadelphia, Pa.
Carl G. Barth, 420 Whitney Avenue, New Haven, Conn.
Earle Buckingham, Professor, Department of Mechanical Engineering, Massachusetts Institute of Technology, Cambridge, Mass.
Ralph E. Flanders, Manager, Jones & Lamson Machine Company, Springfield, Vt.
Arthur M. Greene, Jr., Dean, School of Engineering, Princeton University, Princeton, N. J.
Clarence W. Ham, Professor of Machine Design, University of Illinois, 115 Transportation Building, Urbana, Ill.
Charles H. Logue, *Secretary*, Consulting Engineer, 123 Clarke Street, Syracuse, N. Y.
Fred E. McMullen, Manager Cutter Department, The Gleason Works, Rochester, N. Y.
Edward W. Miller, Chief Engineer, Fellows Gear Shaper Company, Springfield, Vt.
Ernest Wildhaber, 379 Alexander Street, Rochester, N. Y.

teeth, the elasticity of the material would cause a variation in the smoothness of the flow of power.

When these gears are operated under load, this variation in deformation would tend to accelerate and decelerate the gears, and, because of their inertia, would result in an increasing and decreasing tooth load as well as a corresponding variation in the speed of the gears.

The acceleration would take place as the contact left the portion of the profile where the static deformation was greatest to engage that part of the profile where the static deformation was less. This condition would exist when the second tooth of the driving member started to engage the second tooth of the driven member. This acceleration would tend to increase the tooth load on the portion of the profile where the static deformation was less, and act to smooth out the action by increasing the deformation at this portion of the profile. Thus at low speeds where the influence of inertia is less, the variations in velocity would follow very closely the variations in the static deformation, but as the speeds increased and the

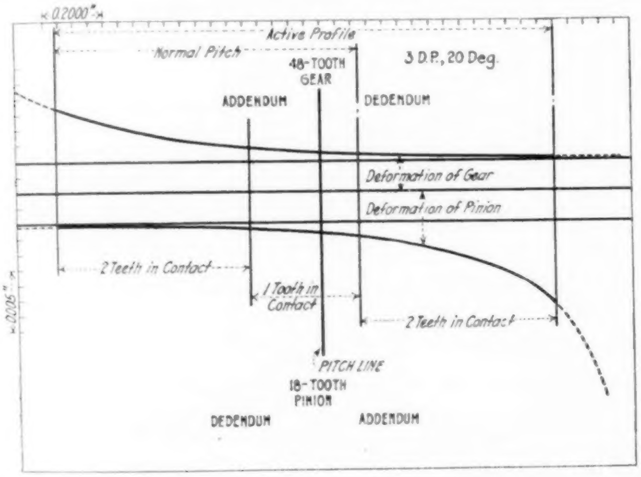

FIG. 2 TOTAL DEFORMATIONS ALONG ACTIVE PROFILES OF TOOTH AND PINION UNDER A STATIC LOAD OF 1000 LB. APPLIED SUCCESSIVELY AT ALL POINTS ALONG SAID ACTIVE PROFILES

effects of inertia were greater, the variations in velocity would become less and less, until a balanced condition was reached where the effects of inertia were sufficient to cause a uniform deformation of the tooth profile at all points of contact. At such a speed the gears would travel at a constant velocity, although a variation in the tooth load would still be present. This variation in the tooth load would be the difference in the static load required to distort all points of the profile equally under the specific conditions of mesh.

STATIC DEFORMATION OF GEAR TEETH

Formulas for the calculation of the deformation of gear teeth are given in a paper entitled The Strength of Gear Teeth, by S. Timoshenko and R. V. Baud, published in MECHANICAL ENGINEERING, November, 1926. These are as follows:

When b = width of strip of contact between two cylinders under load
P = load per inch of face
E = modulus of elasticity of material, and
r_1, r_2 = radii of cylinders in contact,

$$b = 3.04\sqrt{\frac{P}{E} \times \frac{r_1 r_2}{r_1 + r_2}} \quad \ldots\ldots [1]$$

When d_1 = distortion due to compression and
m = Poisson's ratio of the material,

$$d_1 = 2\,\frac{1 - m^2}{E} \times \frac{P}{\pi}\left(\frac{2}{3} + \log\frac{4r_1}{b} + \log\frac{4r_2}{b}\right) \quad \ldots\ldots [2]$$

The radius of curvature on an involute-gear-tooth profile is changing constantly as the diameter changes. However, when a pair of involute gears are meshed, the sum of the radii of curvature $(r_1 + r_2)$ on the mating profiles is constant, and is equal to the center distance multiplied by the sine of the pressure angle. Equations [1] and [2] may be combined and simplified as follows:

$$\log\frac{4r_1}{b} + \log\frac{4r_2}{b} = \log\frac{16r_1r_2}{b^2}$$

$$b^2 = 10.336\left(\frac{P}{E} \times \frac{r_1r_2}{r_1 + r_2}\right)$$

whence

$$\log\frac{16r_1r_2}{b^2} = \log\frac{1.548E(r_1 + r_2)}{P}$$

and

$$d_1 = 2\,\frac{1 - m^2}{E} \times \frac{P}{\pi}\left(\frac{2}{3} + \log\frac{1.548E(r_1 + r_2)}{P}\right) \quad \ldots\ldots [3]$$

It will be seen from Equation [3] that the amount of compression depends upon the sum of the radii of curvature in contact. As this sum is constant on a pair of mating involute-gear-tooth profiles, the deformation due to compression will be constant over the entire active profile as long as the pressure is constant.

When d_2 = deflection due to bending of tooth
L = length of tooth to sharp point
a = distance from sharp point to point of application of the load
h_0 = thickness of tooth at base, and
h = thickness of tooth at point of application of load,

$$d_2 = \frac{12PL^3}{Eh_0^3}\left[\left(\frac{3}{2} - \frac{a}{2L}\right)\left(\frac{a}{L} - 1\right) + \log\frac{L}{a}\right] + \frac{4P(L - a)(1 + m)}{(h + h_0)E} \quad \ldots\ldots [4]$$

Equation [4] is that for a cantilever beam of variable depth. The first term on the right-hand side represents the deflection due to the bending moment, and the second term represents the deflection due to the shearing force.

As a definite example to determine the general nature of these distortions, the static distortion on a pair of 3 d.p. gears of 20 deg. pressure angle has been calculated by means of Equations [3] and [4]. The pinion has 18 teeth while its mating gear has 48 teeth. The material is steel. This gives the following values for Equation [3]:

E = 30,000,000
$r_1 + r_2$ = 3.7622
m = 0.30

P =	500	1000	2000	3000	4000
d_1 =	0.000116	0.000246	0.000519	0.000802	0.001091

It will be noted from the foregoing tabulation that the distortion due to compression is very nearly directly proportional to the load, so that for all practical purposes it may be assumed to vary directly as the load.

An inspection of Equation [4] will show that the amount of bending will also vary directly as the load. We shall therefore calculate the amount of bending due to a static load of 1000 lb. applied at various positions along the tooth profiles by the use of Equation [4]. We have for these gears

	18-tooth pinion	48-tooth gear
L	0.8402	0.9174
h_0	0.6050	0.6901

These deflections are tabulated in Tables 1 and 2.

TABLE 1 BENDING DEFLECTION OF 18-TOOTH, 3-D.P., 20-DEG. STEEL PINION

Position along line of action	a	h	Deflection factor	Deflection, 1000-lb. load
−0.7799	0.6771	0.5770	0.00000002628	0.000026
−0.5294	0.6444	0.5746	0.00000003402	0.000034
−0.2708	0.5884	0.5605	0.00000004815	0.000048
0 (Pitch point)	0.5069	0.5236	0.00000008247	0.000082
0.2882	0.3965	0.4491	0.00000016253	0.000163
0.6013	0.2518	0.3152	0.00000038939	0.000389
0.9452	0.0655	0.0883	0.00000148362	0.001484

The total distortions along the active tooth profiles of this 18-tooth pinion meshing with the 48-tooth gear under a static load of 1000 lb. applied successively along all points of the active profiles of a single pair of teeth are plotted in Fig. 2. Actually, the load

TABLE 2 BENDING DEFLECTION OF 48-TOOTH, 3-D.P., 20-DEG. STEEL GEAR

Position along line of action	a	h	Deflection factor	Deflection, 1000-lb. load
−0.8621	0.8365	0.6607	0.00000001038	0.000010
−0.5805	0.7636	0.6275	0.00000002107	0.000021
−0.2936	0.6797	0.5827	0.00000003844	0.000038
0 (Pitch point)	0.5841	0.5236	0.00000006620	0.000066
0.3010	0.4762	0.4469	0.00000012204	0.000122
0.6134	0.3541	0.3501	0.00000023434	0.000234
0.9303	0.2201	0.2288	0.00000048692	0.000487

will be distributed over two pairs of teeth for part of the action as indicated in Fig. 2. The total deformation is plotted in Fig. 3 to a more enlarged scale, and the resultant distortion when two pairs of teeth are sharing the load is plotted in dotted lines. This resultant is calculated as follows:

When d_a = deformation per unit load on one pair of teeth
d_b = deformation per unit load on second pair of teeth, and
d_r = resultant deformation when two pairs of teeth are dividing the load between them,

then

$$d_r = \frac{d_a d_b}{d_a + d_b} \quad [5]$$

Although the contact along the line of action will cease as soon as the tip of either tooth profile crosses it, the load will not be in-

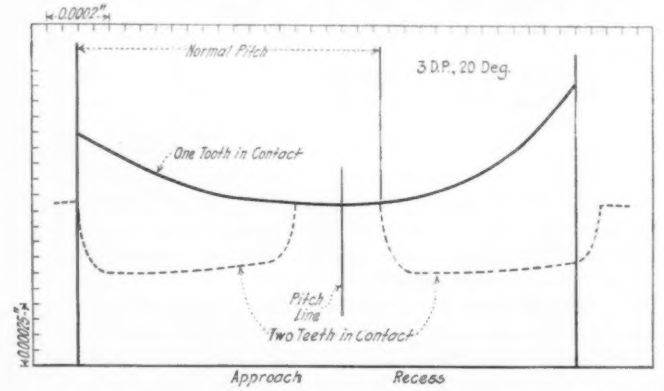

Fig. 3 Total Deformations of Tooth and Pinion of Fig. 2 Plotted to an Enlarged Scale

(The resultant distortion when two pairs of teeth are sharing the load is plotted in dotted lines.)

stantaneously transferred to the pair of teeth remaining in theoretical contact because of the deflection of the profiles. An exact solution of the distance in which this transfer of load is effected would be a very complicated one. We shall therefore use the following approximation for this purpose:

When d_t = amount of distortion when load is transferred to or from a single pair of teeth
D = distance point of contact travels along line of action while load is transferred
A = distance along line of action from pitch point to end of theoretical action
a = radius of base circle of pinion
R_1 = pitch radius of pinion, and
R_2 = pitch radius of gear,

then

$$D = \frac{R_2 a}{R_1 + R_2} \arccos \frac{A - d_t}{A} \quad [6]$$

In the foregoing example we have the following values:

$d_t = 0.00067$, $A = 0.8600$ } At beginning of single tooth contact
$d_t = 0.00066$, $A = 0.7530$ } At end of single tooth contact
$a = 2.819$
$R_1 = 3.000$
$R_2 = 8.000$,

whence $D = 0.0809$ at beginning of single tooth contact
$= 0.0858$ at end of single tooth contact.

These values are used in plotting Fig. 3, which shows the resulting amount of deformation on the profiles of these gear teeth under static conditions as the load is shifted from point to point along the line of action.

Variation in Load Caused by Static Deformation

As noted before, the variation in the amount of deformation on these gear-tooth profiles would cause accelerations and decelerations of the gears when running under load with a corresponding variation in the total load on the teeth. Thus,

when E = difference of deformation in feet
f = applied load in pounds
v = pitch-line velocity in feet per second
m_1, m_2 = effective masses of gears at pitch line
F = maximum tooth load, and
f_1 = additional load caused by acceleration,

$$F = f + f_1 \quad [7]$$

$$f_1 = ma \quad [8]$$

where a = acceleration caused by deformation.

$$S = v_0 t + {}^1/_2 at^2 \quad [9]$$

where S = space in feet,
v_0 = initial velocity in feet per second, and
t = time in seconds.

Whence

$$a = \frac{S - v_0 t}{{}^1/_2 t^2} \quad [10]$$

$$S = D_1 + E \quad [11]$$

where D_1 = distance along pitch line in which load is transferred, in feet.

$$D_1 = \frac{D}{12 \cos \alpha} \quad [12]$$

where D = distance along line of action in which load is transferred, in inches, and
α = pressure angle of gears.

$$t = \frac{D_1}{v_0} \quad [13]$$

whence

$$a = \frac{2Ev_0^2}{D_1^2} \quad [14]$$

$$f_1 = \frac{2mEv_0^2}{D_1^2} \quad [15]$$

Transforming Equation [15] into the following units:

V = pitch-line velocity in feet per minute = $\frac{v_0}{60}$
e = difference of deformation in inches = $12E$, and
D_2 = distance along pitch line in which load is transferred in inches = $12D_1 = D \cos \alpha$,

we have

$$f_1 = \frac{meV^2}{150D_2^2} = \frac{meV^2 \cos^2\alpha}{150D^2} \quad [16]$$

Equations [11] to [16] apply only to rigid solids, that is, considering the static deformations as rigid. With elastic bodies, as the acceleration load increases the difference in deformation will become less, which in turn will reduce the amount of the acceleration load. This acceleration load will be relatively small at low speeds, but will increase rapidly as the speed is increased because of the greater influence of the inertia of the bodies.

Carl G. Barth has pointed out that under these conditions the resulting load f_1 will start out as a parabola at the low speeds and approach a horizontal asymptote at the higher speeds. Thus when $f_1 = C_1V^2$ is the equation of the parabola which would represent the reactions of rigid solids, and $f_1 = C_2$ is the equation of the horizontal asymptote which would represent the influence of elasticity, the resulting force would be given by the equation

$$f_1 = \frac{C_1V^2C_2}{C_1V^2 + C_2} \quad [17]$$

In other words, when it takes less force to accelerate the masses than it does to distort the tooth profiles, the masses will be accelerated more and the distortion will be less. On the other hand, when it takes less force to distort the profiles than it does to accelerate the masses, the distortion will be more and the acceleration less. Equation [16] thus becomes the equation of the parabola which represents the force required to accelerate the masses without considering any further distortion of the tooth profiles.

We must now establish the value for the horizontal asymptote. This will be the load required to distort the second pair of profiles an amount equal to the distortion on the single pair when it is carrying the load. From an inspection of Fig. 3 it will be seen that this load will be somewhat less than the applied load, as the distortion increases as the contact travels away from the pitch line.

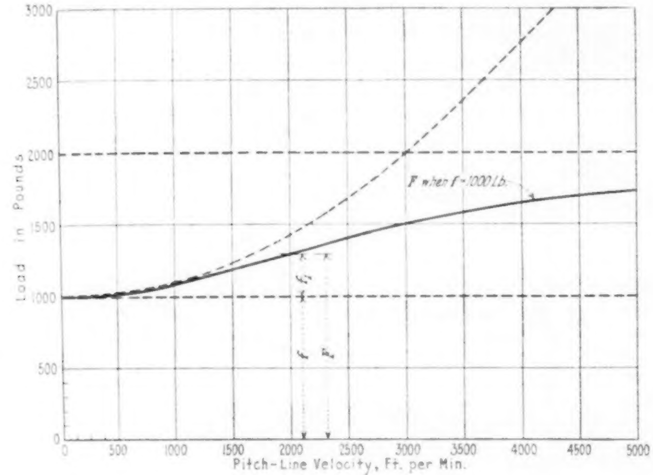

FIG. 4 VALUES OF MAXIMUM TOOTH LOAD F FOR VARIOUS VELOCITIES

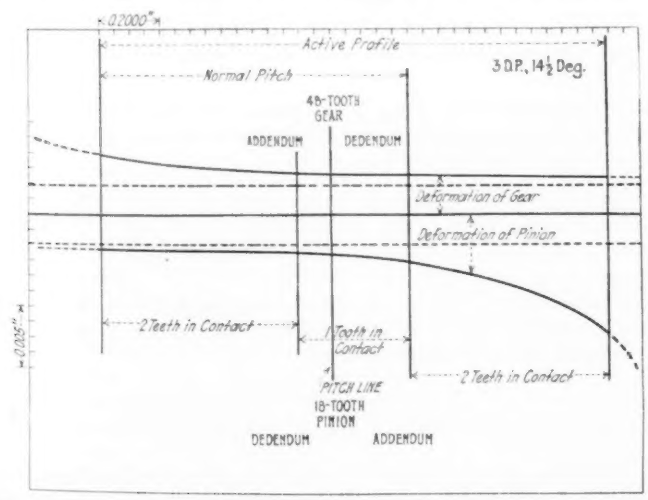

FIG. 5 TOTAL DEFORMATIONS AT VARIOUS POSITIONS ALONG THE LINE OF ACTION

Under the conditions shown in Fig. 3, this load would be about 80 per cent of the applied load. Under other conditions it might be slightly more or slightly less. For the sake of simplicity we can assume that this load will be equal to the applied load, which assumption will cause no material error. The equation of the horizontal asymptote will then become

$$f_1 = f \qquad [18]$$

whence Equation [17] becomes

$$f_1 = \frac{meV^2 \cos^2\alpha\, f}{meV^2 \cos^2\alpha + 150D^2 f} \qquad [19]$$

and Equation [7] becomes

$$F = f\left(1 + \frac{meV^2\cos^2\alpha}{meV^2\cos^2\alpha + 150D^2 f}\right) \qquad [20]$$

When m = effective mass at pitch line of gears
m_1 = effective mass of pinion at pitch line, and
m_2 = effective mass of gear at pitch line,

$$m = \frac{m_1 m_2}{m_1 + m_2} \qquad [21]$$

In the previous example we have, from Fig. 3,

$$f = 1000$$
$$e = 0.000275$$
$$D = 0.0858,$$

whence
$$F = 1000\left(1 + \frac{0.0002428mV^2}{0.0002428mV^2 + 1104.246}\right)$$

Assuming both effective masses equal to 32.2 lb., we have

$$m_1 \text{ and } m_2 = \frac{32.2}{g} = 1, \text{ whence } m = \frac{m_1 m_2}{m_1 + m_2} = \frac{1}{2}$$

and

$$F = 1000\left(1 + \frac{0.0001214V^2}{0.0001214V^2 + 1104.246}\right)$$

Values of F for velocities of from zero to 5000 ft. per min. determined from the foregoing equation are plotted in Fig. 4, together with the parabola representing the reactions of rigid solids and the horizontal asymptote representing the influence of the elasticity.

STATIC DEFORMATION OF A PAIR OF 14½-DEG. GEARS

We shall now consider another example, using 14½-deg. gears instead of 20-deg. gears, and determine the static deformation in the same manner as before. These gears will otherwise be the same as before, which will give us the following values:

$$E = 30{,}000{,}000$$
$$r_1 + r_2 = C\sin\alpha = 2.7542$$
$$m = 0.30$$
$$P = 1000,$$

whence, from Equation [3],

$$d_1 = 0.000240$$

	18-tooth pinion	48-tooth gear
L	0.8965	1.0260
h_0	0.5520	0.6350

The bending deflections at the several positions along the line of action are given in Tables 3 and 4. These are determined with

TABLE 3 BENDING DEFLECTION OF 18-TOOTH, 3-D.P., 14½-DEG. STEEL PINION

Position along line of action	a	h	Deflection factor	Deflection 1000-lb. load
0 (Pitch point)	0.5632	0.5236	0.00000009483	0.000095
0.3058	0.4724	0.4816	0.00000016639	0.000166
0.6032	0.3585	0.4027	0.00000033140	0.000331
0.9256	0.2095	0.2620	0.00000082124	0.000821

TABLE 4 BENDING DEFLECTION OF 48-TOOTH, 3-D.P., 14½-DEG. STEEL GEAR

Position along line of action	a	h	Deflection factor	Deflection 1000-lb. load
0 (Pitch point)	0.6927	0.5236	0.00000007546	0.000075
0.3649	0.5936	0.4737	0.00000012981	0.000130
0.6636	0.5013	0.4186	0.00000021130	0.000211
0.9700	0.3965	0.3472	0.00000036229	0.000362

Equation [4]. These values are also plotted as before in Figs. 5 and 6.

In this example we have

$$\left.\begin{aligned} d_t &= 0.00065 \\ A &= 0.8875 \end{aligned}\right\} \text{At beginning of single tooth contact}$$
$$\left.\begin{aligned} d_t &= 0.00070 \\ A &= 0.7511 \end{aligned}\right\} \text{At end of single tooth contact}$$
$$a = 2.9045$$
$$R_1 = 3.000$$
$$R_2 = 8.000$$

Whence, from Equation [6],

$$D = 0.0809 \text{ at beginning of single tooth contact}$$
$$= 0.0912 \text{ at end of single tooth contact.}$$

Fig. 6 shows the resulting composite deformation of the two pairs of mating teeth as a dotted line as before. The general characteristics here are very similar to those shown in Fig. 3.

Elasticity Form Factor

An elaborate analysis is not always possible for every pair of gears that may be used. We shall therefore examine the foregoing analysis to determine whether or not simple approximations are possible which will shorten these computations and give reasonably accurate results.

We shall first examine Equation [3] for the amount of compression. We have already seen that this distortion is constant over the entire profile of a single pair of teeth, and that it is also practically directly proportional to the intensity of the load. We shall now determine its variation when the tooth profiles are changed.

Using the same values as before except for a change in the value of $(r_1 + r_2)$, we obtain the following results from Equation [3]:

$(r_1 + r_2)$ =	0.94055	1.8811	3.7622	7.5244	15.0488
d_1 =	0.000219	0.000233	0.000246	0.000259	0.000273

It will be seen from the foregoing tabulation that a change in the value of $(r_1 + r_2)$ has so little influence on the amount of compression that it can safely be ignored.

We shall now introduce values for cast iron instead of steel into this equation, which gives us the following result:

$$E = 15{,}000{,}000$$
$$m = 0.27$$
$$r_1 + r_2 = 3.7622$$
$$P = 1000$$

whence $d_1 = 0.000474.$

In this case, when the modulus of elasticity is reduced to one-half, the amount of deformation is very nearly doubled. In other words, this deformation is practically inversely proportional to the modulus of elasticity of the material.

We shall therefore establish the following approximation for the amount of distortion due to compression:

$$d_1 = \frac{7.25P}{E} \qquad [22]$$

We shal' now examine Equation [4], which gives the deformation due to bending. In this case the amount of bending is always directly proportional to P/E. Usually the load is transferred from one tooth to the next when the contact is somewhere near the pitch line of one tooth and at the tip of the other tooth. The maximum deformation will be at the pitch-line area, as the contact is here when a single tooth is carrying the load. We shall therefore consider this deformation at the pitch line. We have from Tables 1, 2, 3, and 4, when the load is 1000 lb.,

	18-tooth, 20-deg.	48-tooth, 20-deg.	18-tooth, $14^1/_2$-deg.	48-tooth, $14^1/_2$-deg.
d_2 =	0.000082	0.000066	0.000095	0.000075
h_0 =	0.605	0.6901	0.552	0.635
L_0 =	0.667	0.667	0.667	0.667
$\frac{L_0}{h_0}$ =	1.1025	0.9665	1.2083	1.0504
$\frac{L_0^3}{h_0^3}$ =	1.3401	0.9028	1.7641	1.1589

where h_0 = thickness of tooth at base
L_0 = height of tooth.

If the effective deflection due to bending were directly proportional to L_0^3/h_0^3, we should have from the foregoing examples when $K = \frac{P}{E} \times \frac{L_0^3}{h_0^3}$ and d_2 = constant $\times K$:

	18-tooth, 20-deg.	48-tooth, 20-deg.	18-tooth, $14^1/_2$-deg.	48-tooth, $14^1/_2$-deg.
d_2 =	1.835K	2.060K	1.615K	1.941K

These constants show considerable variation. Due to the fact that the thickness and length of a gear tooth are very nearly equal, it may be that the Lewis tooth-form factor y may be used as a close approximate measure of the distortion due to bending. Thus we shall let $K = \frac{P}{E} \times \frac{1}{y}$ and d_2 = constant $\times K$. This would give the following:

	18-tooth, 20-deg.	48-tooth, 20-deg.	18-tooth, $14^1/_2$-deg.	48-tooth, $14^1/_2$-deg.
y_2 =	0.098	0.130	0.083	0.112
d_2 =	0.24108K	0.24180K	0.23655K	0.25200K

These constants show less variation than the preceding ones, so we shall tentatively adopt the following approximation:

$$d_2 = \frac{0.242\,P}{y \times E} \qquad [23]$$

Combining Equations [22] and [23], when d = total effective deformation,

$$d = d_1 + d_2 = \frac{P}{E}\left(\frac{0.242}{y} + 7.25\right) \qquad [24]$$

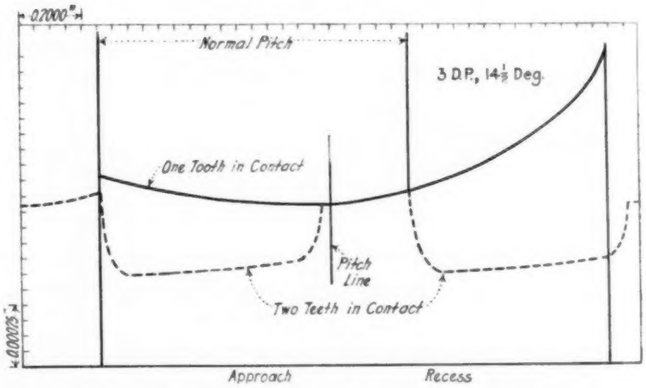

Fig. 6 Total Deformations of Fig. 5 Plotted to an Enlarged Scale
(Resultant composite deformation of two pairs of mating teeth plotted in dotted lines.)

The approximation given by Equation [24] would represent the average effective distortion of one gear tooth meshing with another. Thus when z = elasticity form factor,

$$z = \frac{P}{E \times d} = \frac{1}{\frac{0.242}{y} + 7.25} \qquad [25]$$

When a pair of teeth were engaged we should have, when z_1 and z_2 = elasticity form factors, E_1 and E_2 = moduli of elasticity, and d_t = total amount of deformation,

$$d_t = P\left(\frac{1}{E_1 z_1} + \frac{1}{E_2 z_2}\right) = P\left(\frac{E_1 z_1 + E_2 z_2}{E_1 z_1 \times E_2 z_2}\right) \qquad [26]$$

We shall now use the approximation given by Equation [24] and [25] to check the deformation caused by a load of 1000 lb. on the teeth of the four gears previously used. These equations give us the following results, under which are tabulated those obtained from Equations [3] and [4]:

	18-tooth, 20-deg.	48-tooth, 20-deg.	18-tooth, $14^1/_2$-deg.	48-tooth, $14^1/_2$-deg.
z =	0.1029	0.1097	0.0984	0.1063
d =	0.000322	0.000304	0.000339	0.000314
Original d =	0.000328	0.000312	0.000335	0.000315

It would appear from these examples that the approximations used are sufficiently close for all practical purposes.

Approximation for Distance on Pitch Line to Transfer Load

The next problem is to establish a simple general equation to determine the distance traveled along the pitch line while the load is transferred from one tooth to the next. This will be accomplished by modifying Equation [6]. The distance equivalent to A will be taken as three-fourths of the circular pitch, which will be a fair average. Thus when

p = circular pitch in inches
d_t = total amount of effective distortion
R_1 = pitch radius of pinion
R_2 = pitch radius of gear, and
D = distance along pitch line gears move while load is transferred,

Equation [6] will become

$$D = \frac{R_1R_2}{R_1 + R_2} \arccos \frac{0.75p - d_t}{0.75p} \qquad [27]$$

Approximation to Be Employed for Difference in Static Deformation

The difference in static deformation between a single pair of teeth in contact and two pairs will always be slightly less than one-half the static deformation of a single pair. For the purpose of simplification we shall assume it to be equal to one-half the static deformation on a single pair at the pitch line. Thus when e = difference in static deformation in inches,

$$e = \frac{d_t}{2} \qquad [28]$$

Acceleration Load

With the foregoing approximations we should have for the acceleration load caused by the deformation of the teeth the following modification of Equation [19]:

$$f_1 = \frac{meV^2}{meV^2 + 150D^2f} \qquad [29]$$

and the following modification of Equation [20] for the maximum load:

$$F = f + \frac{meV^2}{meV^2 + 150D^2f} \qquad [30]$$

where F = maximum total load
f = applied load
m = effective mass at pitch line of gears (Eq. [21])
V = pitch-line velocity in feet per minute
D = distance along pitch line in inches (Eq. [27])
e = difference in deformation in inches (Eq. [26] and [28]).

In concluding this analysis of the influence of elasticity on perfect gears, it should be pointed out that this acceleration load would be imposed principally, if not entirely, upon the undeflected tooth profile as it was coming into action, so that with perfect gears the acceleration load due to the deformation probably would not increase the load on a single tooth to an amount greater than the transmitted load. The total load would be increased, but this increased load would be distributed over two pairs of teeth. On the other hand, the reaction from the deceleration load when one pair of teeth is leaving contact would probably be very nearly equal to the acceleration load, and this reaction, or the force required to restore the lost velocity, would be carried by a single pair of teeth in addition to the transmitted load. If this reaction was sufficient to cause the teeth to leave contact, they would come together again with an impact which would impose a heavier additional load than either the acceleration or the reaction from the deceleration. With perfect gears and a constant applied load, however, such an impact is not possible because the additional loads caused by the deformation will always be less than the applied load and hence will not be sufficient to overcome the influence of this applied load. For the sake of simplicity, the maximum value of this additional load at infinite velocity has been taken as equal to the applied load, but actually, as pointed out before, the load at this limiting condition will always be slightly less.

With imperfect gears, however, we must deal with both acceleration loads and impact loads. As the next step in the analysis of the influence of elasticity on gear-tooth loads, we shall therefore consider first the acceleration loads, a certain measure of which has been given by the tests on the gear-testing machine, and then attempt to analyze the impact loads.

Location of and Coal for Central Power Plants

A GREAT deal of misinformation exists among the public generally as to the requirements of a central power station, many people advocating the building of these plants at the mouths of coal mines. "This," in the words of S. A. Taylor, in a paper on the above subject presented at the Second Mid-West Power Conference in Chicago, Ill., February 15 to 18, 1927, "is generally the result of lack of knowledge of the water requirements of a large central power plant." Places where there is an abundance of both coal and good water are very few, he pointed out, and to these two prime factors is added the additional problem of securing this location near a center of distribution of power.

One of the first considerations in the location of a power plant is the matter of current distribution. If for a large city and surrounding communities alone, then it will likely be found more desirable to build at some point where water enough can be secured and if fuel cannot be secured close at hand to secure it from the district where the most desirable coal can be obtained, at the lowest freight rate. If the plant is to serve a large area, it may be possible to locate it where both fuel and water can be obtained, but it may require more power lines to carry the current to the points of consumption. The author had found it more economical in several instances to build the extra-long power lines in order to secure the advantage of plenty of water and coal. Among other points to be considered are delivery and storage of coal, when furnished directly from the mine, and railroad facilities. He urged the use of what may be termed the "deadly parallel" in determining locations, enumerating on one side all the advantages with their costs and on the other the disadvantages with their respective costs.

Regarding the selection of coal, the author commented on the fortunate development of equipment permitting the use of small sizes and powdered coal, enabling the miners to load all coal mined, including the fine coal formerly retained at the mine, and permitting larger production from a given area. Among the many other factors entering into the selection of coal are the following:

1 Where coal is close to the plant and where there would be no freight charge against the coal, arrangements can be made to use a poorer grade profitably, on account of the cheaper cost of coal delivered at the plant. In all cases where good coal can be secured close to the plant this is more desirable to obtain, even at a higher cost per ton or acre. This is also advantageous in that no other transportation than that of the mine has to be depended upon. Regarding the future supply of coal, the author said that in many cases this was based on the maturity of the bonds. That is, enough coal is purchased to supply the needs of the station until the bonds have matured—thirty, forty, or fifty years, as the case may be. Here enters a rather delicate bit of calculation, for while overhead and carrying charges may be determined quite accurately taxes have a way of fluctuating in an annoying manner. Carefully prepared figures will enable the engineer to determine the best course to pursue as to future requirements and location of supply.

2 Where it is necessary to ship the coal into the plant from a distance, the location of the mine giving the best fuel, the transportation charges, the amount and cost of coal available for the future, etc., must be given consideration. In this case more attention must be given to quality than in the first case, for freight rates on good and bad coal are about the same.

3 Generally, deep-mined coal is considered more desirable than strip coal, but he mentioned two tests in particular which showed 10 per cent better results for the strip coal. This was accounted for by the better preparation of the latter.

4 Another item of importance mentioned was the fusion point of the ash, which, taken in connection with furnace temperatures, would determine generally the matter of clinkering.

Bituminous-Coal Preparation in the Eastern Fields

Important Results Obtained Through Coal Preparation—Nature and Source of Ash—Methods of Preparation—Mechanical Cleaning and the Various Types of Cleaning Equipment—General Preparation Conditions in the Various Large Producing Districts

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IT IS OBVIOUSLY impossible, in the space available to go into any detailed description of coal-preparation methods and equipment in each of the eastern bituminous-coal fields. Moreover, there is considerable variation in the size of the different fields from the standpoint of production as well as of area, while the coal-preparation methods used in the larger fields are also in use in the smaller ones.

For our purposes it will be sufficient to consider the smaller fields in conjunction with the larger ones. The author will endeavor, therefore, to give a general picture of the whole subject, without going into details regarding the various fields, their coals, and the preparation methods used.

Before dealing with specific practices in the eastern fields, it will be advisable (1) to discuss coal preparation and what it involves, with a description of the general methods used, and (2) to specify the fields to be included, their location, and their production tonnages.

"Coal must be regarded as a mixture of the true fuel, containing its unavoidable complement of inherent ash, with extraneous incombustible matter." Dr. Lessing, the British coal chemist, who has probably done as much research as any one into the nature and effects in use of coal ash, is responsible for the foregoing quotation. Dr. Lessing adds, "A close study of the valueless portion of coal is one of the urgent needs of fuel research, and the results likely to be obtained from it will be of help in enhancing the value of our most vital source of energy—Coal."

Preparation of coal, its screening and cleaning, involves a study of the different methods and practices used and the principles underlying them; of the ash, its occurrence, geology, nature, and effects in use of its constituents; of its chemistry and physical characteristics, and of plant physiology as related to inherent ash. And this means not only a study of the general subject along the lines indicated, but of their application to each coal to be treated, especially if some method of mechanical cleaning is to be used.

The primary object of coal preparation is to increase the value and uniformity of the product at the lowest cost to the producer and with the least possible amount of waste.

Some of the other important results of efficiently-carried-out coal preparation are—

1 Making possible the satisfactory use of low- or medium-grade fuels by certain industries for purposes in which such fuels had not been previously satisfactory or could be used only with difficulty
2 Conservation of our higher-grade coals
3 Better results in general use
4 Savings in handling and transportation costs.

Coal preparation is generally understood to mean the sizing of coal after mining by the use of suitable stationary or moving screens, and the removal from it of "extraneous" mineral matter, as distinct from that which is inherent; the latter cannot be removed by any known commercial method.

In bituminous coal as shipped, the amount of extraneous ash is generally greater than the amount of inherent ash, the proportion of each varying with different coals. Inherent ash seldom amounts to more than 3 per cent but may exceed 20 per cent; usually it ranges from 2 to 8 per cent. The total ash in the bulk of commercial coal ranges from 6 to 12 per cent, but often runs higher in inferior grades.

The principal differences between the various kinds of coal and their ash constituents are due to the differences in the plant or vegetable materials from which the coal was formed and to the varying agencies, with their relative action, by which these vegetable materials have been altered to the material coal as we know it today.

Ash, What It Is and Whence It Comes

Generally speaking, inherent ash is made up of the mineral constituents of the vegetable matter from which coal originated, together with the depositions of mineral matter from the waters to which the coal has been exposed during its formation.

Extraneous ash, on the other hand, is made up of materials of an entirely different character, the principal ones being shale, clay, bone (really a high-ash coal), iron pyrites, marcasite, calcites (calcium carbonates), or ankerites (ferrous carbonates).

These impurities, found in coal as shipped, come from—

1 The roof or bottom of the coal bed, by which is meant the rocks, usually shales or sandstones, that lie next the bed, above or below, and are to a greater or less degree mined with the coal, the amount depending upon their character and the care used in mining
2 Partings (layers or "dirt bands") or "veins" (usually clay) found in the coal bed
3 Balls, lenses, or flakes of inorganic material, usually iron pyrite or marcasite
4 Thin layers or scales on the surfaces of the coal lumps or particles.

It is also quite likely—in fact, it has been apparently proved—that the coal itself is intersected by many very small cracks or fissures that are filled with calcium or ferrous carbonates.

From the foregoing it will be seen that the preparation of coal is not only an important subject, but one involving considerable study and a wide knowledge of chemistry, as well as of mechanics and economics.

To the consumer, preparation of the coal he uses is one of the most important elements in coal production, and there is no question involved in coal purchase, supply, and use that is of greater importance to those having charge of its purchase and use or to those engaged in the design and operation of plants using coal as a fuel.

Coal preparation, its methods and their efficient use, is involved in every phase of production: cutting and shooting of the coal in the mine; its handling from the face to the furnace and the ash dump; mining, transportation, use, cost of plant operation, cost of manufactured product, whether heat, electricity, gas, iron and steel, bread and ice cream, motor cars, or boots and shoes—preparation affects to a greater or less degree each operation and each product of manufacture, as well as the heating of our homes and offices.

There is another important result of efficient coal preparation that is not perhaps generally considered, namely, its part in making a fuel product of uniform character and quality.

The consumer wants for his use a coal of unvarying (within possible limits) quality and character. Modern preparation methods, if efficiently handled, will go far toward giving him such a coal, and will often overcome possible variations in heating value as between different coals.

A coal of uniform quality means uniformly better use results, and this applies to every use, whether domestic or industrial. It is especially true with respect to coal used by the carbonization industries, which consume almost one-fifth of our total bituminous production.

Ernest Prochaska, in his book on Coal Washing, lays down the following maxims as applied to coal preparation:

1 The preparation of coal shall, by the cleaning of the raw material and the production of suitable and well-screened sizes, secure a maximum price per ton of output

[1] Consulting Engineer. Mem. A.S.M.E.

Presented at a meeting of the Metropolitan Section of the A.S.M.E., New York, February 11, 1927.

2 To arrive at this result three points must be kept in view: (*a*) highest possible purity of coal; (*b*) smallest possible loss of coal; (*c*) small cost of production

3 As the foregoing three demands are conflicting, it will be necessary for the proper and economical installation of a preparation plant to find in each case the best relation between the three factors.

Preparation of coal naturally begins in the mine at the face where the coal is cut. In fact, it may be said to start with the solid coal bed, as the method of preparation to be used at any mine is largely dependent upon the structure of the bed, the character of the rocks above and below it, and the number, size, and nature of the partings, veins, balls, flakes, and lenses found in the bed itself.

In the cleaning of the coal or the removal of the extraneous impurities that make up the larger part of the total mineral content, sizing or screening is largely involved, particularly with the smaller sizes and in mechanical cleaning, whether of the wet or dry type.

Methods of Preparation

Coal-preparation methods in general are classified as—

1 By hand
2 Mechanical.

Each method includes two main phases:

1 Sizing or screening
2 Cleaning, or the removal of the extraneous impurities.

Sizing is accomplished by running the coal as it comes from the mine (or sometimes after crushing), (*a*) over stationary screens which may be either bars or perforations, (*b*) over shaking or vibrating screens of various types and sizes, or (*c*) through revolving screens.

In the early days of the industry (*a*) stationary screens were almost universally used, and are still in use at smaller mines, but today most of the large mine tipples are equipped with shaking screens; revolving screens are not widely used, for various reasons.

Sizing of coal was first brought about by the demand for lumpy coal and the feeling on the part of the consumers (still in evidence) that the slack or small coal was, as it is still often called, "dirt."

The number of sizes made, beginning with lump and slack, has gradually risen until today as many as twelve different sizes are made and shipped in the Illinois fields, including run-of-mine, or unscreened coal as it comes from the mine.

Generally, however, in the eastern fields, there are four or five standard sizes—lump, egg, nut, and slack, together with the run-of-mine. Here again there is considerable variation in practices as between different fields and coals; such variations depending largely upon the uses to which the coal is to be put, whether for steam making, coking, gas manufacture, or domestic heating.

In addition to the commercial sizes, the small coal, generally that below $2^1/_2$ in., must be given a further separation, either by screening or by crushing and screening, if some form of mechanical cleaning is used, although in the old days of the hand-operated jigs used in washing, no size classification of coal was thought necessary. Modern methods, however, call for sizing before washing as well as afterward.

While the basic theory of screening coal is simple, commercial application develops complications, such as the shape of particles, the proportions of small and large, the moisture, the shape of the screen openings, the slope and motion of the screen, and the velocity of movement over it.

In the earliest days of the bituminous-coal industry in this country, screening was done underground by the use of rakes or forks. Then followed the use of stationary bar screens, still used at many of the smaller mines. Shaking or revolving screens were probably first used in the bituminus fields as early as 1880. In Great Britain the practice of screening coals at the mine probably began about 1775.

The whole problem of sizing or classification is one that must be solved for each coal, taking into consideration all of the factors involved, the structure of the coal, the nature of the impurities, the method of washing or cleaning, the use to which the coal is to be put, and the cost of equipment and operation.

In the eastern bituminous-coal fields screening practices vary considerably, both with respect to the number of commercial sizes made and the types of screens used; the latter depending to a large extent upon whether the coal is cleaned mechanically or by hand, and what type of mechanical cleaning or washing is used.

Cleaning of Coal

The removal of extraneous impurities in the preparation of coal for shipment begins in the mines where the miners and loaders are required (not always with success) to throw out the impurities before loading the coal into mine cars. In the early days of the industry this was the only place where any attempt was made to clean the coal. Then followed in this country, in addition to the underground cleaning, cleaning by hand in the railroad car as it was being loaded, a method still in use at many smaller mines, and capable of giving excellent results with some coals. For about 20 years this was the only cleaning system used in the Pocahontas and New River fields of West Virginia. This method was superseded at the larger operations by the use of picking tables or conveyors, over which the coal, generally sized first, passes and the impurities are removed by hand.

The smaller sizes are not cleaned at all where the coal is sufficiently free from impurities to need no further preparation, as is the case with many of our best coals.

This brings us to another method not exactly new, but developed to its present status in recent years.

Mechanical Cleaning of coal or washing it to remove the impurities was first brought about by the demands of the coke-making industry for cleaner small coal, the latter being better adapted than the large sizes for coke production. A patent for the washing of coal by means of different specific gravities was taken out in England by Charles Cowper in 1849, although some method of washing had been in use in Germany before that date.

In this country the earliest use of a coal-washing method seems to have been in Illinois in 1884, but in the last fifteen years, and especially in the last five, its development has been quite rapid especially in certain districts where mechanical cleaning of the small coal came to be a necessity.

All mechanical washing of coal, whether by dry or wet process, is based on the differences in specific gravity between coal and the various impurities it may contain as mined.

The various methods of mechanical washing may be divided into two classes:

1 Wet washing, where the medium is water or water and some other material
2 Dry washing, where air is the usual medium for effecting the separation between the coal and impurities.

Wet Washing. Under this would come apparatus for washing coal by difference in the falling velocities through water between coal and the extraneous impurities. These would include froth, current washers, jigs, classifiers, stationary tables, shaking tables, slime tables, thickeners, and clarifiers.

The processes in which separation is effected by the use of a liquid heavier than coal and lighter than the impurities, include the use of solutions of salts in water and suspension in water of solid materials heavier than coal. These processes require the application of energy in order to maintain the suspension of such solid particles in water, giving such suspension the fluid properties of a liquid with a density greater than coal and less than that of the impurities.

There are still other methods with suitable apparatus whose operation is based upon molecular or atomic properties. These include the several types of oil, froth, and other wet flotation processes, together with certain magnetic and electrostatic types, the former being either a wet or dry suspension and the latter essentially a dry process.

In the eastern bituminous-coal fields, washing of coal has been practiced to a greater or less extent for some 30 years especially in certain districts where the small coal is high in ash content and difficult to market and use in its raw state.

There are several different types of wet washers now in use in the eastern fields which are either of new design or are adaptations of older methods.

Dry Washing. The operation of these processes and apparatus is based upon differences in specific gravity, in friction or resistance, and in energy and movement of moving particles.

Table 1 shows the 1923 production of washed coal in the eastern states and the percentage of cleaned coal to total state output. No later figures are available. For the entire country the total production of washed coal in 1923 was 20,140,385 net tons, or 3.6 per cent of the total soft-coal production.

Preparation and Its Effect on Ash-Fusion Temperature and the Clinkering of Coals

First let it be remembered that coal ash is not a simple substance but a very complex one, not only in the number of its constituents but also in the varying relations between such constituents.

The seven or eight principal constituents of coal ash may and do vary in amounts even in different samples from the same mine and bed. This being a fact, we are not surprised to find that different samples or shipments from the same mine vary as much as 700 deg. fahr. in their determined ash-fusing temperatures. And this is supplemented by the additional fact that different lots of coal from one mine will, when burned under boilers, vary considerably in their clinker-forming action, even when burned under practically identical conditions.

With such possible variations among the constituents of coal ash, in ash-fusing characteristics, and between the inherent ash and the extraneous impurities, the effect of preparation must be determined for each individual coal.

Changes in preparation may either increase or decrease ash fusibility, depending upon the nature of the inherent ash and of the extraneous impurities.

Mechanical Loading in Mines, Effect on Preparation

The introduction of mechanical loaders at the mines, a movement that is growing rapidly and must be reckoned with, has added greatly to the interest taken in mechanical preparation. It is generally admitted that loading machines, if operated to capacity, give no opportunity for removing impurities from the coal in the mine by hand. For this reason all cleaning of coal must be done at the tipple, which in many cases makes the use of some mechanical method imperative for successful operation and a clean product.

Types of Mechanical Cleaning Equipment

In the eastern fields there are in operation practically all of the commercially developed types of mechanical cleaning devices from the older jig and trough washers to the more modern wet and dry cleaning and flotation processes.

During the past two years, and especially in 1926, much progress has been made in the construction of new mechanical cleaning plants and in the commercial progress of new devices, as well as in the equipment of mines with screens and hand-cleaning arrangements.

In addition to the various types and sizes of screens and hand-picking tables used in each of the eastern fields, there are in operation or under construction a number of plants for cleaning coal mechanically.

A list of bituminous washing and dry-cleaning plants and additions in 1926 appeared in *Coal Age*, January 27, compiled by Thomas Fraser of the Pennsylvania Geological Survey.

The following eastern field installations are taken from that list:

Name	Location	Capacity, tons per hr.	Type of washer
East Broad Top Coal Co.	Union, Pa.	400	Chance
Berwind-White Coal Co.	Windber, Pa.	400	Arms air table
Algoma Coal & Coke Co.	Algoma, W. Va.	200	Arms air table
Elk River Coal & Lumber Co.	Widen, W. Va.	160	Arms air table
Winding Gulf Collieries Co.	Winding Gulf, W. Va.	50	Arms air table
Winding Gulf Collieries Co.	Winding Gulf, W. Va.	25	Arms air table
Winding Gulf Collieries Co.	Winding Gulf, W. Va.	25	Arms air table
Pittsburgh Coal Co.[1]	Library, Pa.	325	Arms separator
American Coal Co. (Addition)	McComas, W. Va.		American separator
Davis Coal & Coke Co.[1]	Boswell, Pa.		Air tables
American Coal Co.	Widemouth, W. Va.		Jeffrey Robinson
American Coal Co.	Winona, W. Va.		Jeffrey Robinson
H. J. Patterson Coal Co.	Arista, W. Va.	30–40	Jigs
Southern Coal & Coke Co.	Boothton, Ala.		Deister-Overstrom
Black Diamond Coal Co.	Johns, Ala.		Deister-Overstrom
Black Diamond Coal Co.	Birmingham, Ala.	40–50	Elmore

[1] Under construction, started in 1926.

TABLE 1 COAL PRODUCTION IN EASTERN STATES IN 1923

State	Cleaned coal, net tons	Percentage of cleaned coal to total state output
Alabama	12,285,695	60.1
Georgia	24,161	32.0
Kentucky	306,436	0.7
Pennsylvania	2,496,843	1.5
Tennessee	390,275	6.5
Virginia	261,346	2.2
West Virginia	1,144,693	1.1

Of the various mechanical cleaning devices in operation or being installed in the eastern fields, the following brief descriptions are given.

Chance Sand-Flotation Process. This is a flotation process in which coal, of less specific gravity than the separation-fluid mass, floats and the heavier impurities sink. With this process it is possible to make separations which closely reproduce the usual laboratory float-and-sink tests. This process, first used in the anthracite field, has been in successful commercial operation in the central Pennsylvania field for over a year, it is reported.

Arms Air Table and Dry-Cleaning Process. This includes Arms screens and dry-cleaning pneumatic tables, the latter perforated. A continuous current of air is supplied to the tables by a fan and coal is fed to them from the Arms screen, over which the coal is propelled by the shaking motion at a uniform rate. The air currents stratify the coal, which is lifted by the air and goes over the side, while the heavier refuse remains on the surface and is moved to the end of the deck by the shaking motion.

American Pneumatic Coal Separator. This is a perforated metal deck through which air is forced by a centrifugal fan and which has a lateral and longitudinal slope. The top of this deck contains a number of riffles which are higher at one end than at the other. The deck has a reciprocating motion in the direction of the riffles. Coal is automatically fed to the separator at its highest point and is stratified on the deck by the air. The escaping air is sufficient to raise the clean coal above the riffles and the latter travels by the aid of gravity to the low point of the deck. The higher-density materials settle on the deck surface and by a reciprocating motion pass between the riffles to the high side of the deck and are there discharged. As the products of varying densities are stratified they are also zoned at different points on the deck. The separator will treat all sizes of coal from $2^1/_2$ in. diameter to fine dust; the coal is first screened into a number of sizes depending upon the character of the product to be treated.

The Peale Pneumo-Gravity Process. This process, developed by Peale, Peacock & Kerr, is in operation at several of their mines in central Pennsylvania. It is claimed that it will handle all sizes up to and including 3-in. lump.

Trent Process. Two plants using the Trent process have been erected, one in central Pennsylvania and the other in northern West Virginia. This process involves the mixing of pulverized coal with oil and water, and effects a marked reduction in mineral content of the coal treated.

Rheolaveur Process. A type of wet washer recently brought to this country, the Rheolaveur, has had marked success in the anthracite field of Pennsylvania, but has not as yet entered the eastern bituminous fields, as far as actual operation is concerned. It is, however, well adapted for preparing bituminous coal and is widely used in Belgium and France where there are several hundred installations; there are also about 20 in operation in Great Britain, all handling bituminous coal.

The author will not attempt to describe the various types of wet washers that have been and are in successful commercial use in the eastern fields, notably in Alabama. They are, among others, the Montgomery, Elmore, Stewart, Luhrig, Faust, and the James types.

The Eastern Bituminous-Coal Fields

As classified and described by the 1923 U. S. Coal Commission, these include all producing fields in Pennsylvania, West Virginia, Virginia, Eastern Kentucky, Tennessee, Maryland, and Alabama.

The producing districts in each state are listed at the top of the following page.

The number of mines and production for 1925 in the eastern bituminous fields are given in Table 2; 1926 figures are not yet available.

Pennsylvania
- Pittsburgh (including Westmoreland)
- Connellsville
- Westmoreland-Ligonier
- Freeport
- Butler-Mercer
- Blossburg
- Broad Top
- Somerset
- Central Pennsylvania

Maryland and West Virginia
- Maryland-Potomac (Upper Potomac)

Virginia
- Southwestern Virginia
- Clinch Valley
- Semi-Anthracite
- Richmond Basin

Kentucky
- Northeast Kentucky
- Hazard
- Harlan

Alabama
- Pratt
- Big Seam (Warrior)
- Cahaba

West Virginia
- Fairmont
- Panhandle-Pittsburgh No. 8 (in Ohio)
- Putnam County
- Kenova
- Thacker
- Tug River
- Pocahontas
- Winding Gulf
- New River
- Kanawha
- Coal River
- Logan
- Coal and Coke
- Preston County
- Taylor County
- Junior-Philippi
- Gauley

Tennessee
- Rockwood
- Fentress

Kentucky-Tennessee
- Southern Appalachian
- Jellico

General Preparation Conditions in the Various Large Producing Districts

Alabama. Alabama is said to have been the first state to experiment actively with coal washers. It was also the first state to install large-scale washeries and a larger proportion of washed coal is sent out than in any other state, about 62 per cent of its total production being delivered to washeries for treatment. Most of the mines not equipped with washeries have screens and hand-cleaning equipment. As far as known, all of the washeries are of the wet type.

Eastern Kentucky. In Eastern Kentucky the most continuous growth of coal production has been experienced in the Harlan County, Hazard, and Big Sandy districts. The respective production of these districts in 1914 and 1925 was—

	1914 Net tons	1925 Net tons
Big Sandy (or Northeastern)	2,970,000	10,600,000
Hazard	820,000	10,900,000
Harlan County	24,000	11,500,000
Total of above	3,817,000	33,000,000

While the bulk of growth in production in the Big Sandy region has come from plants mining by-product and other special-purpose fuel, the Hazard field has expanded very largely due to the output of high-grade domestic coals. Although the same is true to a large extent in Harlan County, the increase has been more evenly distributed with large increases in the production of captive coal, with the main growth at commercial plants mining quality grades of domestic coal.

Coal in the Big Sandy River district has not had to meet such strict demands in preparation to find an outlet and so the equipment is not generally so flexible as to sizing as in the other two sections.

While there are relatively few tipples equipped for making four sizes of coal simultaneously, in the Harlan and Hazard fields there are a majority of the commercial plants which can make more than three sizes of coal, and almost one-half of them can make more than four sizes when demand arises.

Of the commercial mines in these two districts, the percentages of tipples for various preparation advantages are approximately as follows:

	Harlan County District, per cent	Hazard District, per cent
Able to make more than 3 sizes	75	68
Able to make more than 4 sizes	46	47
Equipped with shaking screens	82	79
Equipped with picking tables	49	52
Equipped with loading booms	69	58

The above does not reflect the portion of the total respective outputs subject to the indicated preparation; the percentages of production supplied with such facilities would be considerably larger in each instance.

Pennsylvania—Pittsburgh and Thick Freeport Districts. The Connellsville region, including the Latrobe district, has probably the smallest percentage of preparation of any district. The coal is soft and friable and is usually shipped as mine run. In some few cases bar screens are used, more particularly in the Latrobe section where the coal becomes a little harder.

In the Greensburg basin both gravity or bar screens and mechanical screening equipment are used, with the necessary picking tables and loading booms. In some cases the coal is washed before coking. The three sizes produced in this district are lump, mine run, and slack.

Both mechanical screening and gravity screening are used in the Irwin basin, where the coal becomes harder than in the other two localities mentioned, and the sizes shipped are lump, nut, slack, and mine run.

This also applies to the Youghiogheny Gas district, and to that portion of the Youghiogheny Gas district extending across the Monongahela River into Washington County. In this district there has been considerable attention given of late to the proper cutting and shooting of the coal at the face in order to increase the percentage of lump. Various types of shaker screens, as well as bar or gravity screens using either $^3/_4$-in. or $1^1/_4$-in. spacings are employed; picking tables and loading booms are also becoming more generally used in order to not only decrease degradation but to eliminate as much ash as possible.

TABLE 2 PRODUCTION OF MINES IN EASTERN BITUMINOUS FIELDS IN 1925

State	Name of field	No. of mines	Production, net tons
Pennsylvania	Pittsburgh	274	25,121,000
	Connellsville	211	35,010,000
	Westmoreland-Ligonier	115	13,939,000
	Freeport	89	8,089,000
	Somerset		6,730,000
	Central Penna., West		
	Central Penna., Middle	178	8,866,000
	Central Penna., East	685	32,107,000
	Small adjacent fields (3)	146	4,158,000
Maryland-W. Va.	Maryland-Potomac	125	4,305,000
West Virginia	Fairmont	192	19,433,000
W. Va.-Ohio	Panhandle-Pittsburgh No. 8	186	21,697,000
W. Va.-Kentucky	Thacker	67	7,965,000
	Kenova	23	2,908,000
W. Va.-Virginia	Pocahontas	96	19,904,000
West Virginia	Winding Gulf	71	9,316,000
	New River	117	10,907,000
	Kanawha & Coal River	195	16,535,000
	Logan	113	19,255,000
	Taylor Co., Junior		
	Philippi & Gauley	119	5,347,000
	Small fields (4)	90	3,842,000
Virginia	Southwestern Virginia	75	8,618,000
	Clinch Valley	27	2,417,000
Kentucky	Northeast Kentucky	166	14,061,000
	Hazard	93	7,625,000
	Harlan	71	11,440,000
Kentucky-Tenn.	Southern Appalachian	140	6,573,000
	Jellico	32	1,009,000
Tennessee	Rockwood-Soddy	47	2,122,000
	Fentress	7	751,000
Alabama	Big Seam Group	46	6,022,000
	Cahaba Group	102	5,800,000
	Pratt Group	92	8,248,000

Inasmuch as most of the mines in this district produce the best grade of gas coal and therefore obtain higher prices than the other districts, more expensive equipment can be installed than in some of the other districts. The sizes shipped are block, lump, nut, slack, and mine run.

A great many mines in the Irwin and Youghiogheny Gas districts are on the retreat, and are therefore producing coal with a greater percentage of impurities at the present time than was formerly the case. They are having much difficulty holding the high-class markets for this particular reason.

In the Thin Vein Youghiogheny Gas districts and the Pan Handle Steam Coal districts, practically the same methods are used as in the Irwin and Youghiogheny districts.

A recent check-up of the tipple equipment of the coal mines in the Pittsburgh Freight district, which includes practically all mines in southwestern Pennsylvania except those in the upper and lower Connellsville regions, showed as follows:

Number of mines	373
Mines with mechanical screening equipment	66
Mines with bar or gravity screens	160
Mines with no screening equipment	154
Mines with picking tables	72
Mines with loading booms	60
Mines with coal crushers	11
Mines with coal washers	7

In the central Pennsylvania districts there has been great improvement in the past two or three years in the matter of coal preparation, although there is still a large tonnage of coal loaded at the old-style tipples where coal is cleaned in the railroad cars. Many of the larger mines, however, use hand-picking tables and screens. There have also been several installations of mechanical cleaning devices, including the Chance process, the Arms dry cleaner, and the Peale pneumo-gravity process. Other installations of mechanical cleaning devices are in prospect in this field.

Tennessee. In Tennessee there is not so heavy an investment in preparation equipment as in the adjacent fields, but the larger mines are generally equipped with screens and picking tables, although many still load run-of-mine coal only. There are some six or eight wet washeries in the state.

West Virginia. The small percentage of washed coal shipped from West Virginia mines does not mean a lack of interest in coal preparation in the fields of that state. On the contrary, the operators in all of the principal fields pay a great deal of attention to preparation of their coals, and there is no field in the East that shows greater material evidences of interest in the matter. While there are comparatively few washeries in any of the West Virginia fields, some of the best-equipped preparation plants in the country are found there. The natural quality of many of the West Virginia coals has made mechanical cleaning unnecessary, up to the present time.

In the Fairmont and other northern West Virginia and Maryland districts, preparation equipment is almost entirely limited to screens and picking tables of the usual types, while there is still considerable coal loaded which is cleaned in the railroad car.

There is no field in the country where more attention has been paid to coal preparation than the Pocahontas. Practically all the mines are now equipped with modern tables, well equipped for screening and preparation. There are also 21 wet washeries and 2 dry cleaning plants in this district, with others in process of installation.

Preparation in the Winding Gulf and New River fields is largely limited to screens and picking tables, with which nearly all of the mines are equipped. There are, however, three dry-cleaning plants and two wet washeries in operation, and a great deal of attention is given to both inside and outside preparation.

Mines in the Kanawha field are generally well equipped with screens and picking tables, although there still are a number of mines where coal is cleaned in the car. There are five washeries in the field.

Operators in the Logan field have always given close attention to preparation of their coals. There is hardly a mine in the district which is not equipped with up-to-date screening devices and picking tables. So far it has not been found necessary to wash any of the Logan County coals.

The author's thought in preparing the foregoing paper has been to give those interested in the use of coal, as distinct from its production, general information regarding coal preparation, its problems and methods, together with a picture of today's conditions, as they exist in the eastern bituminous-coal fields.

In conclusion he would say that every engineer whose work has to do with the use of coal, whether in plant design or in plant operation, should have some knowledge of coal-preparation practices and processes, and their relation to furnace, oven, and retort operation, as well as to transportation and economics.

* * *

Commenting on the paper, Samuel W. Balch wrote that any one who had sauntered along a mountain stream must have noticed how thoroughly it sorted out the various materials in its way. A lump of coal and a piece of slate he said, differed from each other in three physical ways: First, the coal was lighter; second it was more or less cubical or globular and presented more surface to a horizontally flowing stream; third, friction, it would roll and move more easily than slate. The Rheolaveur process utilized these three differences simultaneously, and preliminary close sizing was not required. Water rushed down an inclined trough or launder. Near the head, the run of the mine was discharged. Below was a trap box opening into the bottom of the trough through which the slate was discharged. Further along was a second trap box where the lumps of adhering slate and coal were taken out. In the ordinary run-of-mine there was not enough of this intermediate material for the proper functioning of the apparatus and it was therefore discharged into an elevator boot, returned to the head of the trough and again washed down and circulated so that a thick bed was maintained into which the run-of-mine was introduced, the slate working down and the coal free from slate working up. These middlings imparted to the stream the property of a liquid of intermediate specific gravity so that the separation was effective. Separation into sizes as required for the market was made by screens at the lower end of the trough.

The Usefulness of Vacua

STRIVE as they may, scientists have been unable to attain a vacuum wherein a cubic inch includes fewer molecules than there are people in the world. Even so they have succeeded in removing 99.999,999,990 per cent of the gas. In other words, only one of every 10,000,000,000 molecules remains; yet there are 40,000,000,000 molecules in every cubic inch; the population of the earth is estimated at less than 2,000,000,000.

Across the broad girth of America, from New York to San Francisco, a four days' journey by train, thirty hours by swiftest airplane, imagine a great belt of fine sand a thousand feet, or nearly a fifth of a mile, in width and ten feet deep. Its length, from coast to coast, would be more than 2500 miles. Then imagine it suddenly reduced to a line, so slender as to be almost invisible, just two grains broad and one grain deep.

That is a graphic illustration of how completely a modern Coolidge X-ray tube is exhausted of air by the high-efficiency Langmuir condensation pump in the research laboratory of the General Electric Company. The great beach with its millions of millions of grains of sand is symbolic of the cubic inch of air at atmospheric pressure, if each of its molecules were enlarged to the size of a grain of sand one one-hundredth of an inch in diameter, and the line two grains broad and one grain deep represents the almost complete vacuum obtainable with the Langmuir pump. No vacuum known to science is absolutely complete.

The swiftness with which the air is drawn out is equally marvelous; the Langmuir pump accomplishes this in just two seconds.

Materially, there are countless thousands of molecules in the highest vacuum attained; there is also endless interest and utility. In fact, the American public is paying more than a million dollars a week for glass-contained vacua.

Ability to create even partial vacua in enclosed spaces has been of great use. It has made possible suction pumps, thermometers, incandescent electric lamps and many improved physical and chemical processes, and has increased the efficiency of steam engines and turbines.

At night we see largely by the aid of vacuum lamps. By means of other vacuum lamps (X-ray tubes) we can also see through opaque bodies. Our transcontinental wired telephony is possible through vacuum tubes, which, in various forms, also permit our radio broadcasting and radio reception from the most remote stations. One of the latest achievements of science, the transmission of photographs by wire or wireless, incorporates still another vacuum tube, the photoelectric cell.

The workman who keeps his drink hot or cold in a thermos bottle is indebted to Sir James Dewar's application of the vacuum, but the scientist is still more indebted to it. Our steam-power plants, including turbines, also owe their success to vacua.

Vacua are of different types. For the steam turbine, a rough vacuum is sufficient—about an inch of mercury. Incandescent lamps formerly required the best vacua available, but there are now other commercial vacua necessarily quite superior, such as those for X-ray and radio tubes, and for the newly developed cathode ray tube.

It appears evident, or at least probable, that the condition which humans call vacuum is the natural state throughout the vastness of interstellar space, and some of the stars—millions of miles in diameter and countless millions of miles away from us—are but superheated gases, less dense than the residual gas in the most nearly perfect vacuum tube we know.—From *Research Narratives*, vol. 7, no. 3, March, 1927.

What Is Known About the Effect of Smoke on Health

Investigations on the Physiological Effect of Smoke—Proposed Plan for Further Investigation—Physiological Effects of Carbon Particles in Smoke—Physiological Effects of Other Dusts in Smoke

By W. C. White,[1] Washington, D. C.

SMOKE is the most visible pollution of the air. In the last fifty years our cities of greatest industrial activity in steel and railroads have turned day into night, and night into intense darkness. There are many reasons for demanding the control of smoke, but we have so far failed to make the most powerful of all reasons—that smoke has a baneful influence on the health of those who must daily breathe it—so convincing that all will clamor for its suppression.

Investigations on the Physiological Effects of Smoke

For over two hundred years, since the memorable book of Ramazzini, the subject of the effect of smoke on health has intermittently come before thoughtful minds in medicine and engineering. Almost one hundred years passed from the time of Ramazzini until Aronson, Laennec, and Gregory made it a subject of importance, and not until 1860 did Traube prove that carbon in the air left its tracings in the lungs, pleuræ, and glands of those who breathed such air. This is within the lifetime of some who read these words.

It was an easy mental step from the proof that carbon particles in breathed air were deposited in the lungs, pleuræ, and lymph glands, to the suggestion that these particles must influence disease processes in these same organs. Tuberculosis, pneumonia, bronchitis, the most prominent diseases of the lungs, were naturally chosen to demonstrate this belief.

Fifty more years passed by before this suggestion was portrayed so graphically that the general public and reformers began to take an interest in it. Ascher, between 1905 and 1911, aroused such interest through a very elaborate series of statistical studies and some experimental work. Despite the fact that the methods he used for obtaining individual figures were so variable that the additions of these were not convincing, and that later more careful studies proved some of Ascher's conclusions faulty, he nevertheless commanded attention for the subject. A great deal of experimental work on animals followed his work but, as is well known, there is always grave danger in transferring the results of animal work to conclusions regarding man. The long-observed difference in the lungs of donkeys and lungs of smaller animals during a lifetime in coal mines, as pointed out by Watkins-Pitchford, is suggestive of the danger that lies in this field. The respiratory apparatus, the susceptibility to disease, the resistance of the nose, and the food habits, all provide variants so great as to open the door on the questions under consideration.

In 1914 a group of men undertook a study of the smoke conditions in Pittsburgh. This was made possible by a fund of $10,000 provided by R. B. Mellon. Pittsburgh provides in exaggerated form a constant natural environment and atmosphere of the type that should give the answer to the questions considered in this paper. It is an ideal place to pursue the study of the influence of smoke on diseases of the lungs. Any conclusion drawn from such studies however, to have influence, must be so graphic as to overcome other great factors. As every one knows, smoke means active business, profit to owners, and high wages to workers. In 1907 when first the author went to Pittsburgh there was depression in the mills and every one in this period longed for more smoke as an evidence of prosperity.

Pittsburgh presents two striking facts. It has a low tuberculosis death rate year after year. On the other hand, it has the highest constant pneumonia death rate of any community in the world. Furthermore, in analyzing the pneumonia death rate of the city by wards, it became clear that the denser the smoke content of the air, the higher the pneumonia death rate. With respect to tuberculosis, on the contrary, the density of smoke seemed to bear no relation to the tuberculosis death rate. If one could reason safely from this study, one would say that smoke had no influence on tuberculosis but had a tremendous influence on acute lung diseases. This is probably near the truth, but there is not yet enough evidence to make such a statement. The analysis recently carried out by Dr. Thompson of the U. S. Public Health Service indicates that there is an industrial health hazard in work in the steel mills that has a definite relation to the high pneumonia mortality in Pittsburgh.

Pneumonia as a basis of argument for the prevention of smoke has never been very successful. This is undoubtedly due in large part to the lack of definite knowledge as to the effects of smoke on health in causing or promoting diseases. In general, wherever efforts are being made by communities with a smoke nuisance to prevent smoke, such efforts are based on consideration of discomfort rather than health. If it can be definitely proved, as the above study indicates, that smoke causes a high death rate from pneumonia, although the public has never appreciated the importance of acute respiratory diseases due especially to their after effects, this will be a powerful argument for prevention of smoke.

Before explaining the experimental work that justifies the statement that the breathing of a smoke-laden atmosphere has no demonstrable influence on tuberculosis but is of importance as a baneful factor in pneumonia and other acute diseases of the nose, throat, and lungs, the author would present for consideration a bit of evidence that he has tried as a measuring rod upon the basis of supply and demand. If acute diseases of the upper air passages, such as colds, laryngitis, rhinitis, pharyngitis, and bronchitis are more prevalent in a smoky atmosphere, one would expect to find the number and wealth of physicians who devote themselves exclusively to the treatment of these diseases to be greater in Pittsburgh than in cities of the same size but without a smoky atmosphere. The Register of American Physicians shows that Pittsburgh, for a population of 580,000, has 41 nose and throat specialists, while Baltimore has 40 for a population of 730,000. These figures show that Pittsburgh as a market provides for a similar number of such practitioners, although Baltimore is a city one-fourth larger. Those who have indulged in the luxury of sprays, swabs, and much surgery know that this is a great added expense to the citizens of any community.

Perhaps more convincing than this are the data that resulted from a study which the author made of eight city squares in Pittsburgh compared with the rest of the city. The eight squares chosen were in one of the low-river-level wards on the South Side in the mill district. The outstanding observation in this district was the prevalence of acute respiratory diseases; children with running noses, running ears and all the attendant sequelæ of a constantly infected respiratory tract. As a health hazard this cannot be overestimated, and it is a convincing argument against smoke if it can be shown definitely that the smoke is the primary factor in causing the conditions observed.

As part of the Pittsburgh study the work of Klotz, Haythorn, and Holman gives sound scientific basis for the contention that smoke is a large factor in the prevalence of acute respiratory diseases, and all the suffering that follows these infections. Their observations have been verified both before and since by many other workers, but especially by Mavrogordato, whose careful studies merit special attention.

Such limited evidence is indicative but not conclusive. Here

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again we must conclude that we have nothing sufficiently convincing to leave no doubts in the public mind for the contention that smoke must be controlled from the point of view of its influence on public health.

Proposed Plan for Further Investigation

Therefore, the author would suggest that a practical step be inaugurated—that the properly correlated agencies join with the city of Pittsburgh in a coöperative plan of research to determine definitely the effect of smoke upon health.

Pittsburgh is suggested as an experiment center, because in every way it is suitable. Its high hills and low river levels with its wards distributed on high and low levels, differing often by elevations of 400 feet, lend themselves to such a study. It is one of the smokiest cities. Work already has been done there to suggest basic lines of such research, and the Bureau of Mines has there its finest experiment station.

It is possible in this paper only to summarize the present knowledge without ascribing it to any particular work. Some of it has been verified; all of that presented in this paper, the author feels sure, has sufficient foundation to serve for discussion.

Physiological Effects of Carbon Particles in Smoke

Carbon particles suspended in the air we breathe may lodge upon the surface of the breathing apparatus from its beginning at the nose to the smallest air vesicle in the depths of the lung. Most of this is blown out on the handkerchief or swallowed in nasal or throat secretions. That which is swallowed is mostly passed out by the bowel, while that passing the natural barriers in the nose acts as foreign matter, often causing irritation, as every one who has had a cinder in the eye knows. The immediate response to such an irritation is a pouring out of fluid in its neighborhood. This may be the result either of a change in the pH of the neighborhood, which pure carbon particles have the capacity to arouse, or it may be due to an actual change brought about by substances adsorbed on the carbon particles. Whatever the explanation, the exudate occurs. The direct result is that any pathogenic organism gaining entrance to the same respiratory tract in the following of the carbon, finds a medium ready at hand for its rapid multiplication.

One such organism makes two and so on about every twenty minutes, so that a dose capable of destruction for any natural immunity of an individual may arise in a few hours. This is in strict accordance with facts. The direct result of the spread of common colds following association with a sufferer is well known. This is probably more rapid in a smoky atmosphere than elsewhere. The organism that produces rhinitis or bronchitis in one individual may cause pneumonia in another, and vice versa. Concerning most of these questions we do not know enough to speak with authority.

Any carbon particles which pass beyond the surface of the body to its interior must have transportation, for they have no means of locomotion of their own. There is one great carrier cell that transports carbon, or probably any other invader of the body from the outside. This cell, known today as the monocyte, is one of the great, restless, wandering cells. Ordinarily it performs a beneficial function in its round of activity in the body from spleen, lymph glands, bone marrow, and omentum through the liver to the lungs, for it transports and burns fat. But it has the stupid habit of picking up undesirable passengers and carrying them from the air tubes of the lungs to the inside of the body. Some of the passengers which it carries, like carbon, it can get rid of at its various stopping places. Others, like the tubercle bacillus, it seems unable to unload. The pneumococcus, the gonococcus, the streptococcus, and others like them, seem to have the power to destroy the cell, but the monocyte apparently can whistle for aid and then a mob of polynuclear cells of wholly different origin and chemical activity come to its assistance. Until we know the oxidation and reduction mechanisms and other relationships of these cells, we shall probably not understand the processes of their relation to disease processes.

Carbon particles picked up by the monocyte are deposited in definite stopping places in the pleura and in the lymph glands and in interlobar trabeculæ. If enough carbon particles are deposited in these places there is a traffic jam, and when later, as in pneumonia, the main highways must be clear to take care of the dissipation of the exudate which accompanies this malady, there is complete demoralization of traffic. Klotz and Haythorn have actually shown that this occurs. Pneumonia does not kill in Pittsburgh in the acute stage as in other cities. But in the later stages when resolution is under way and recovery should follow, the patient cannot recover because a blockade of the ordinary channels for the dissipation of such exudate has resulted from the deposit of carbon in these routes and the litter which this occasions. The best of the known recent work, however, would seem to prove beyond a doubt that none of these changes occur unless the carbon dust is accompanied by siliceous particles of dust.

The same cellular and exudate changes are brought about by carbon and silica dust when accompanied by infections with the tubercle bacillus, but here for some unexplained reason the exudate may undergo the fibrillar colloidizing known to pathologists as fibrosis. This seems to be quite a different process from the fibrous change which occurs in the same exudate where the physical characteristics of a granular type are present. Fibrosis has been, however, the accredited animal protection against the spread of tubercle bacilli, and as a consequence of the increase of fibrous tissue brought about by silica dust and carbon this has been thought to have an inhibitive influence on the spread of tuberculosis. This, however, as is well known, is in quite sharp contrast to the evil influence which we know silica alone has on the tuberculous process in those living constantly in an atmosphere containing siliceous dust. It will be seen, therefore, that our present experimental knowledge is insufficient to explain our impressions that smoke should be suppressed for its beneficial influence on tuberculosis and its evil influence on pneumonia, leaving us still the observed facts that in a constant smoky atmosphere, such as Pittsburgh, the evidence is abundant that there is this wide difference in the dangers accompanying such an atmosphere.

All discussion on the antiseptic properties of smoke seems futile to the author, for there is not the slightest evidence of its activity in this way within the confines of the animal body, although Holman has shown that soot either by physical or chemical attributes is germicidal for certain micro-organisms in the test tube.

The supposition that carbon increases the incidence of cancer is also probably unfounded. The cancer death rate for cities in the United States is highest in San Francisco, one of the choice climates of the world. Pittsburgh, smoky in the extreme, on the contrary, has a comparatively low death rate from cancer. These two facts would indicate that a belief that smoke increases the incidence of cancer can easily be refuted.

Physiological Effects of Other Dusts in Smoke

The dusts associated with carbon emitted from our smokestacks probably are a great deal more important than carbon. Of them we know practically nothing; carbon as the visible suspension so overshadows every other particle. Yet in a day when catalysts are a dominant factor in chemical activity, it is conceivable that small traces of other substances may be of infinitely greater importance. Of one dust, silica, we know a great deal. Long exposure to it stands alone as the one proved baneful influence from dust to which man is exposed. It not only is capable of destroying him and other animals through a series of definite pathological changes which it produces, but it is the one substance which we know favors the tuberculous process both in man and in experimental animals. (Sayers, Gardner, Mavrogordato.) It does this possibly by some chemical process inaugurated by minute amounts over a long period of exposure. Mavrogordato likens its activity to a protective surface glazing, such as eggs undergo by coating with water glass. It seems almost certain that carbon unaccompanied by siliceous dust is not a permanent deposit in the lung, for it is easily expelled without the chemical coöperation of silica particles.

We can demonstrate the influence of minute doses of manganese, iron, arsenic, and other allotropic and polyvalent substances on plant, animal, and microscopic life. It would seem, therefore, that similar influences occur by exposure over long periods to minute amounts of the same substances used in our industrial processes and passed on to the air we breathe through the exhausts of our furnaces. All of these substances can reach our internal chemical

human machinery by transport in the bodies of the monocytes or by direct absorption in solution from our body surfaces, and they may readily be responsible for both physical changes and emotional waves in our population. But, as already pointed out, we know practically nothing of this subject.

The author would therefore repeat that, with Pittsburgh as the experiment center, a carefully planned study ought to be carried on to gain for us the knowledge which we lack today. When we have obtained this knowledge we shall be able to speak with authority not only about the pollution of the air with carbon, but also about all the dusts that may now or in the future pollute the air we breathe.

Discussion of Papers Presented at Session on Smoke Abatement

THREE papers were presented at the Session on Smoke Abatement, which was held under the auspices of the Fuels Division on the afternoon of December 7, 1926, O. P. Hood, Chairman of the Division, presiding. The first paper, by H. B. Meller,[2] dealt with the subject Smoke Abatement, Its Effects and Its Limitations. It appeared in full in the Mid-November, 1926, issue of MECHANICAL ENGINEERING, page 1275. The second paper on the program discussed the problem from the point of view of the physician. This contribution, prepared by W. C. White, appears in the present issue on the immediately preceding pages under the title, What Is Known About the Effect of Smoke on Health. The last paper to be presented was by Osborn Monnett[3] and dealt with the Present Status of the Smoke Problem. It also appeared in the Mid-November, 1926, issue of MECHANICAL ENGINEERING, page 1284.

SMOKE ABATEMENT, ITS EFFECTS AND LIMITATIONS

In the discussion which followed the presentation of Mr. Meller's paper the written comments of Dr. Carey J. Vaux[4] presented the physician's viewpoint.

It had long been appreciated, Dr. Vaux said, that continued irritation of the mucous membrane of the respiratory tract through inhalation of sharp-cornered particles of dust was a predisposing factor in the development of pulmonary tuberculosis, and it was strongly suspected that the great incidence of acute respiratory diseases had a predisposition in the same manner.

Pneumonia, he claimed, was by far the most prevalent and most fatal of all acute infectious diseases. In 1925 in the United States, 108,700 people died of pneumonia, and in proportion to population the number of pneumonia deaths was 50 per cent greater in the cities than in the country districts.

About 75 per cent of all cases of pneumonia were secondary to the acute infectious diseases designated popularly as "common cold," "flu" or "grippe," and, while these conditions were infectious, they, as in primary pneumonia, had predisposing factors as causes leading up to the disease in the individual.

Dr. E. R. Weidlein,[5] in his written comments, said that in Europe it had been found that few cities were so affected as London, for fogs were less frequent, but some continental cities were partially or almost completely surrounded by large factories which operated day and night. Professor Trillat of the Pasteur Institute, he said, ascribed to the foggy days of spring and autumn, when droplets of moisture were suspended in the air and formed a heavy layer a short distance above the ground, the seasonal recrudescence of infectious diseases in large cities.

Chemical analysis of smoke particles, according to medical specialists, had shown the presence not only of unconsumed carbon but also various products due to imperfect combustion in stoves and fireplaces; namely, sulphurous acid, chlorine, ammonia, and various hydrocarbons, which gave rise to permanent irritation of the mucous membranes of the respiratory passages and of the ocular apparatus. The smoke particles with sharp edges (cinders) sometimes caused traumatism of the bronchial mucosa, which might become in turn a portal of entry for the tubercle bacillus. Furthermore, he pointed out, smoke lessened the health-promoting value of sunlight by screening out ultra-violet rays.

Many cities, he said, had endeavored to bring about more complete combustion in stoves, fireplaces, and furnaces. Various devices had been installed in factory furnaces and chimneys to secure better combustion, but much less had been done for the suppression of smoke from dwellings. In France, M. Breton, director of the Bureau of Inventions, had recently decided, at the request of the Conésil Général de la Seine, to inaugurate a prize contest for the discovery of a system that would afford more complete combustion of coal in domestic stoves and furnaces. In addition, an effort was being made to provide a procedure by which the heat ordinarily wasted in factories could be carried to homes near by.

Andrew A. Bato,[6] also in writing, said that Mr. Meller, as well as other authorities, never failed to emphasize the importance of the fireman's work in connection with smoke abatement, and yet no proposition had been made so far to attack the smoke problem from that angle by offering to the firemen a compulsory course in firing with the least possible smoke. To illustrate the meaning of his criticism, he referred to the work of the Prussian Government. The Prussian Department for Commerce and Industry, after having studied the question for eight years by experimenting in governmental plants, had appropriated in 1902 a sum of 40,000 marks annually for conducting courses for firemen, first for two years as a trial, and as the courses proved to be a success, permanently. He explained that the course was given by a government engineer, who traveled from one industrial center to another and gave a two weeks' course in every place, assisted by a master fireman giving practical demonstrations. After a time these courses became compulsory, inasmuch as a man in order to obtain a full fireman's license had first to work in a boiler house as a fireman's helper for two years, then attend the course and obtain a certificate of successful attendance, whereupon he was admitted to the regular examination for a fireman's license. In other parts of Germany the Societies for Boiler Inspection, i.e., societies of plant owners, etc., and first of all the Hamburg Society for the Operation of Fires and Smoke Abatement, had aroused successfully the interests of firemen in the smoke question, in some cases more than expected, as for instance the Firemen's Trade Union of the Hamburg District, which ruled that its members should work only in plants in which they had a chance to earn bonuses for economical and smokeless firing.

Under existing conditions, he pointed out, firemen in hand-fired plants represented by no means the highest type of skilled labor and they needed every stimulation to become interested in their work beyond keeping the proper pressures and water levels.

Another point which he considered to be of importance was the investigation of the atmosphere as to its contents, not of solid matter alone, but also of acrid fumes, especially sulphur dioxide and trioxide. To obtain data concerning this point the municipal government of Karlsbad, the well-known health resort in Czechoslovakia, ordered the suspension of pieces of cloth impregnated with alcalic baryta at various points in the famous pine forests, where the trees were injured by the fumes from the celebrated china factories. After several months of exposure the pieces of cloth were examined for the SO_2 and SO_3 absorbed by the baryta, and it was found that there was a close connection between the amount of these gases and the degree of decay of the trees.

He said that there was no practical way known to burn coal and to prevent at the same time the SO_2 and SO_3 from getting into the atmosphere, but it was believed that the smoke acted as a carrier of these gases in preventing their quick diffusion and concentrating their harmful action to a small area, a theory which seemed to him to be plausible but still to be proved.

Written comment by H. W. Clark[7] touched upon the author's reference to the ordinance limitations of the production of dense smoke, in which he expressed the belief that after smoke abatement had been in vogue for some time in a given community, the two-minute limitation in fifteen was better than the customary six minutes in sixty.

[2] Bureau Chief, Bureau of Smoke Regulation, Pittsburgh, Pa.

[3] Consulting Engineer, Chicago, Ill.

[4] Director, Department of Public Health, City of Pittsburgh, Pa. Died April 15, 1927.

[5] Director, Mellon Institute, Pittsburgh, Pa.

[6] Mechanical Designer, Public Service Production Co., Newark, N. J. Assoc-Mem. A.S.M.E.

[7] Mechanical Engineering Lecturer, University of Utah, Salt Lake City. Jun. A.S.M.E.

In Salt Lake City, he said, they had been working under an ordinance in which the prohibiting clause read as follows: "The emission of dense smoke (No. 3 or greater of Ringelmann Chart) . . . . for a period of one minute except for period or periods aggregating not to exceed six minutes in any one hour, during which period or periods the fire box, or boxes are being cleaned, or a new fire, or fires are being built therein, is hereby declared a nuisance."

They had never felt any handicap in their enforcement work as a result of not having a shorter permissible period of dense smoke, he declared. Salt Lake City was small enough so that an observer located on the highest building could see and record the smoke produced in all plants in the city, so it was not necessary to depend upon the ground men to report violations, except where stacks were hidden from view by intervening buildings. He was convinced that for their conditions the psychological effect of having an ordinance that was considered liberal, and under which they had always been able to effect convictions, even on appeals, more than offset the possible amount of smoke that might legally be produced.

The Salt Lake City ordinance contained no clause which exempted private residences from the provisions of the ordinance, and specifically provided that all residence heating plants must be installed under permit, and must comply with necessary chimney regulations that would result in good draft.

The fuel used for domestic consumption, he explained, was bituminous coal containing 40 to 45 per cent volatile matter, and for the past two or three years the private-residence chimneys had produced far more smoke than all other sources combined. It had been impossible up to that time, however, to get sufficient money in their budget to begin instruction of householders and supervision of the smoke produced from the domestic chimney, but when the funds became available they would be able to go ahead with this work without any additional legislation.

He agreed with the author that compliance secured through coöperation was very much more valuable to the city than when forced. He felt that any city that attempted to obtain smoke abatement solely through prosecution for producing smoke was foredoomed to failure.

He argued that the successful smoke-abatement department was one which had one or more competent technical engineers, capable of analyzing the plant owner's problem, advising changes to be made, and showing the owner how he could soon save the cost of the required alterations through the reduced cost of operating his plant.

Railway roundhouses located in cities presented the most difficult railway problem, particularly where low-volatile coals were not available for part of the fuel used in firing up and conditioning locomotives for the road. Most roundhouses had inadequate draft conditions for the firing-up period. This situation had been helped somewhat by inducing railroads to provide auxiliary blowers in the locomotive stacks, operated from steam connections in adjacent stalls. Even this arrangement proved inadequate for the larger engines. He felt that the only complete solution of the roundhouse problem would come through the use of the motor- or turbine-driven exhaust fan.

He was convinced that cities must come to the idea of spending more money for smoke-abatement work. It was worth its cost. He also felt that part of the problem before the Society was to assist in the formation of a correct public opinion on this vital subject.

John D. Riggs[8] also writing, expressed a feeling that the author's statement that anti-smoke laws had been framed for the purpose of eliminating only a part of the air-pollution evil, and that part the less grave and less menacing to health, stated the case very mildly; for certainly the health of an individual was more important than the cleanliness of the garments worn. It seemed to him that the Ringelmann chart had already played too important a role in city ordinances.

The blackness, of smoke he explained, was due almost entirely to the free carbon it contained, and not to the poisonous elements, although the free carbon and a considerable portion of the objectionable elements had a way of clinging together when both were present in the chimney, and had a greater tendency to settle to the ground than when less free carbon was present. By making the furnaces most efficient and the smoke entirely transparent, there was still a considerable percentage of CO_2 present which, after it had absorbed moisture from the air, became considerably heavier than air and tended to settle to the valleys and low places.

J. F. Barkley,[9] commenting on the statement that "the Bureau of Mines tells us that it is necessary that approximately half the air (the primary) should be fed through the grates and half (the secondary) over the top of the fire," pointed out that without proper amplification, wrong interpretations leading to improper conclusions or applications would arise. This statement, he explained, referred to a principle of combustion in connection with a perfect fuel bed, and this was seldom obtained continuously in practice.

Referring to the fact that in Pittsburgh the law used the Ringelmann chart as a basis, he said that in the city of Washington the law merely stated that there should be no emission of dense or thick, black or gray smoke at any time. This virtually limited the use of coal to that of a very low volatility, which happily was available to the District of Columbia. In working with the various Government plants in Washington, he said he had found that not only a few per cent less volatile matter in a coal would help, but that coals of the same volatile content acted quite differently in this respect. In the actual use, while carrying the same boiler load, some coals would give off their volatile at rates much different from others. The color of the volatile might be brown or black, which would affect the appearance of the smoke. If, through careless firing, a goodly mass of smoke once started in a furnace, it would get through the best arrangements of baffles or arches to the stack, a fact which turned attention back to the old familiar human element.

J. W. Hogg[10] wrote that it was interesting to learn that ash and iron oxide discharged from a stack were perhaps more injurious than the tar products in smoke. He believed that the foremost argument against smoke and matter discharged from the stacks should be presented from a health standpoint.

Reference was made to a recent drive by the officials of Philadelphia against smoke, during which the editors of several papers, with the best of intentions, wrote glaring editorials on the tremendous waste incurred when a stack gave forth smoke.

Not long before that Mr. Hogg had installed a CO_2 recorder in a plant where the engineer and fireman could, with a medium amount of effort, carry the load on the plant with one boiler when they got their fire in good condition. About 11 per cent CO_2 was recorded, because the boiler was small and the setting low, but it was necessary to make smoke.

In an effort to avoid the smoke, the operators tried to coke the coal in the front of the grate and carry a thin fire in the rear. No smoke appeared, but it was necessary to put the spare boiler into service. The CO_2 recorder showed the reason immediately, only between 3 and 6 per cent being recorded.

There were stokers on the market which would burn coal efficiently, using a small amount of excess air without making smoke, he explained, but these were usually installed in plants where artificial draft was employed, with the consequent discharge of matter more injurious than smoke.

One could not question the fact that the discharge of smoke and other matter even more harmful should not be tolerated, but one could not say on seeing a smoky stack that the boiler was being operated inefficiently, he added.

John Hunter,[11] relating his experiences with smoke abatement in a small city, said that he had first insisted on having a mechanical engineer as an inspector. His first move had been against the industrial plants, since in small cities they were usually the most apparent offenders. During his term as engineer for the city of Asheville, N. C., his work on smoke abatement was complicated by the use of high-volatile West Virginia coal, anywhere between seven and forty per. cent. To get the correct amount of air over the fire to bring about proper combustion, jets and cast-iron blocks were provided within the furnaces to induce the divided streams of

[8] Designer, Indianapolis, Ind. Mem. A.S.M.E.

[9] Fuel Engineer, U. S. Bureau of Mines, Pittsburgh, Pa.

[10] Sales and Service Engineer, Republic Flow Meters Co., Philadelphia, Pa. Mem. A.S.M.E.

[11] Mechanical Engineer, St. Louis, Mo. Mem. A.S.M.E.

air to combine with the volatiles as they were distilled at lower temperatures.

The problem of the large heating plant proved rather difficult, the apartment buildings, hotels and large buildings requiring considerable attention. The speaker urged closer coöperation between architects and engineers in the arrangement of boiler systems so that some means would be provided for eliminating smoke.

Mr. Meller, in his closure, said that as in the paper itself, the discussion emphasized that the health menace of air pollution made the damage to buildings, fabrics, etc., great as it was, sink into comparative insignificance. Dr. Vaux cited pneumonia in particular; it was his belief that solids in the atmosphere, by irritating the mucous membrane, constituted a predisposing factor in causes leading to diseases of the respiratory tract. In addition to the direct action of the solids was their effect in causation of fogs and in screening out the ultra-violet rays of the sun.

Mr. Clark and others mentioned the private-dwelling nuisance. During the heating season, unquestionably this type of plant was responsible for most of the annoyance, whether regulation by ordinance was attempted or not. While most of the smoke was caused by improper firing, the householder was not always entirely to blame. Looking at some of the large groups of medium size and small dwellings being erected in our cities; noting the types of boilers or furnaces that were being installed, without any provision for smokeless operation; and considering the inexperience and oftentimes carelessness of the persons doing the firing, it was scarcely to be wondered at that this type of plant supplied the major part of the winter nuisance. In small apartment houses, the practice of having one janitor take care of several houses, including the boilers, added to the aggravation.

Dr. Weidlein reported an effort to provide a procedure by which waste heat from factories could be carried to homes nearby. Such utilization would not only result in reducing the volumes of solids emitted, by lessening the number of operative stacks from private dwellings, but would effect a substantial economy in fuel.

Mr. Bato's suggestion that firemen be licensed was good, but unless the plan included small apartment houses and private dwellings, it would not be properly effective. High-pressure plants had licensed engineers to supervise the work of the firemen, and the larger low-pressure plants had firemen in attendance at all times. It was the spasmodic firing in small plants that was responsible for much of the trouble. Besides, there was the criticism that the dirt evil would not be affected materially. A system of education for fireman would be a benefit, but, unfortunately, city budgets made no allowance for such expenditures; consequently, smoke inspectors had to do the best they could by individual instruction as the case required. A bonus system for good firing was an incentive, and had in a few cases where tried in Pittsburgh made cleaner stacks and more contented firemen.

The most recent addition to the rules governing boiler and furnace installations in Pittsburgh required some provision by which the fireman might know whether or not his stack was smoking. If direct vision to the top of the stack was impossible, as it was usually, an outside mirror could in some cases be placed; otherwise an indicator in the breeching was required. A simple one which served adequately consists of two pieces of stovepipe, each about fifteen inches long, one on each side of the breeching, in such position and at such an angle that the fireman, from the boiler-room floor, might easily see directly through the center line of the pipes, and, of course, across the breeching. At the near end of the pipe nearest the fireman was a piece of ground glass, and at the far end of the other pipe was an electric-light bulb; the ends of the pipes capped to prevent leakage, with caps removable to permit cleaning the glass and the bulb. When the breeching was clear of smoke, light was transmitted clearly to the ground glass; as the breeching passed smoke, the light was obscured, the degree of obstruction directly proportional to the density (in color) of the gas stream. Thus the fireman might know without going outside (which he would not do) just what his stack was showing.

Mr. Clark mentioned that railway roundhouses located in cities presented the most difficult railway problem, particularly where low-volatile coals were not available. The former practice in Pittsburgh had been to use the blower in the locomotive stack, operated from the house steam line. On one road motor-driven exhaust fans had been installed, with very satisfactory results, and it was expected that all roundhouses in the city would provide similar equipment. While, partly because of its accessibility, low-volatile coal was used for firing up, it had been demonstrated that, with the use of the oil burner for lighting off, it was possible to fire up and condition a locomotive for the road, with high-volatile coal only, without dense smoke; it took a little longer, that was all.

Mr. Riggs and Mr. Barkley were of the opinion that the Ringlemann Chart had already played too important a role in city ordinances. Probably the reason for this was that the average person would believe only what he saw, and he saw black smoke coming from stacks. As has been mentioned, investigations had had for their purpose the determination of *all* polluting solids and fumes, while legislation had been toward the suppression of *dense smoke* only. It would seem that making minor changes was more or less begging the question. The air-pollution evil was evident to those who had given it any thought; the next step was, as indicated by Mr. Clark, to have formed a proper public opinion. In this the Society could and should assist.

In the work of smoke abatement as now carried on, the city official was expected only to see that means were provided to keep the stack clear as required by law. As Mr. Hogg stated, a clear stack might mean a very low percentage of CO_2, because of an undue excess of air; in fact, this often was the case. On the other hand, a plant operating at high efficiency might slip over the smoke line occasionally. In neither case did the law consider whether or how much damage was being done by stack emissions which did not color the gas stream. As a matter of fact, other things being equal, the stack carrying the greater excess of air carried more and larger solid particles into the atmosphere.

While considerable was known about air pollution and its evils, it remained to determine more definitely, by scientific survey, just what was emitted under various fuel, firing, and load conditions with different types of boilers or furnaces, what happened to it after its emission, and what harm it actually did. The study was far from complete. The physician, the physicist, the chemist, and the engineer must coöperate. Meantime, much could be done to ameliorate conditions by a close coöperation between architects and engineers, as suggested by Mr. Hunter.

Effect of Smoke on Health

The paper by Dr. White was presented at this point, following which the Chairman asked for discussion, urging those present to confine their remarks to the physician's side of the question. He agreed with the author in the matter of further research being very desirable and that it should have the hearty support of all engineers.

Discussion between Dr. Jerome Meyers[12] and Dr. White as to the effect of carbon particles on the death rate from cancer brought out the fact that there was really no foundation for the statement sometimes made that this disease was caused by the presence of the particles in the air. It seemed that the offending particles might be products of the very high-temperature tars formed in the combustion of coal, probably something as infinitesimal as vitamines in food. Dr. Meyers did not believe that combustion products were the complete cause of cancer, but he felt that they contributed to its increase.

In reply to a question from Prof. A. R. Acheson[13] as to whether the investigation should be made by the doctor or the engineer, Dr. White replied that both should work on the problem. There were phases that were purely engineering and others that were purely medical.

Commenting on Dr. White's paper,[14] Wm. G. Christy wrote that soot, consisting of finely divided carbon particles, acted as a mechanical carrier of sulphur dioxide, which readily combined with moisture in the atmosphere, forming H_2SO_3, sulphurous acid. The H_2SO_3 in turn picked up another atom of oxygen from the air, producing H_2SO_4. This sulphuric acid was carried along by the soot and caused great damage to metal work, buildings, decorations,

[12] Bureau of Industrial Hygiene, New York City Board of Health.

[13] Professor of Mechanical Engineering, Syracuse University, Syracuse, N. Y. Mem. A.S.M.E.

[14] Executive Secretary, Citizens Smoke Abatement League, St. Louis, Mo. Mem. A.S.M.E.

clothing, etc. It also had an irritating effect on the mucous membrane of the air passages and respiratory organs.

Attention was called to numerous ways of arousing public opinion, among them being the newspapers, which were usually interested and ready to help; a wide-awake publicity department; the scores of civic, professional, social, community, and commercial organizations and clubs willing to give valuable assistance. In any American city having a smoke nuisance there would be found many organizations and individuals who were greatly interested in smoke abatement from a civic standpoint. However, care must be exercised to prevent some of these well-meaning men and women from advocating drastic legislation or high-priced smokeless fuels as solutions of the smoke problem.

There had been sporadic smoke-abatement movements in many American cities, he said, but some of these campaigns had achieved small if any results. One outstanding reason for the failure of these movements, was the lack of coördination of efforts of those interested.

Campaigns must be based on the proper knowledge of combustion and other engineering matters involved. The director of the campaign should be a competent mechanical engineer. To get the coöperation of all the varied interests involved, to get the offenders to do something and to keep the campaign going on an even keel, required a high type of diplomacy and tact.

The opinion was expressed that in many of the cities which had smoke nuisances there was a wonderful chance for the local sections of the Society to take the leadership in a smoke-abatement campaign.

Present Status of the Smoke Problem

Following the presentation of the paper by Mr. Monnett, H. W. Clark[15] offered a written discussion in which he heartily agreed with the author that the real problem in smoke abatement was the small heating plant.

In the small plant, although the waste was there just the same, it was not so apparent nor so readily found.

A group of 22 apartment houses which were shown to be the worst smokers were carefully studied in 1921. These buildings had an average consumption of 1550 lb. of coal per 1000 cu. ft. of volume per year. A similar group of 70 apartments which were studied the following summer, and which were not such bad smokers, had an average consumption of 1261 lb. of coal per 1000 cu. ft. per year. The group producing the most smoke burned 23 per cent more fuel than the group that produced the least smoke.

Another example of the waste of fuel by smoke and possible saving by applying modern smoke-abatement methods was illustrated in the data obtained on a group of 13 stoker-fired heating plants. Before the plants were altered to decrease smoke, and operating crews instructed and supervised, this group of plants produced 18,815 minutes of dense smoke during the winter season; after alteration and instruction and supervision of the crews, these plants produced only 1100 minutes of dense smoke during the winter season, a reduction of 94.2 per cent. This resulted in an average saving of 22.2 per cent in the amount of fuel consumed.

Another group of 77 hand-fired heating plants, having steel boilers, were equipped with smokeless furnaces and the firemen instructed in proper firing, with the result that the smoke was reduced 91.2 per cent and the coal consumption 16.4 per cent. In Salt Lake City, he said, tangible savings in fuel equaling $2^1/_2$ times the cost of operating the smoke-abatement department had been shown.

Regarding the author's suggestion that $50,000 per year per million population be set as a minimum figure for effective smoke-abatement work, Professor Clark felt that this proportionate amount of money might be all right for cities of over one million in population, but was inadequate for smaller cities. As an illustration, he said that Salt Lake City had spent more money in proportion to its size than any other American city, yet the amount spent had been inadequate to cover all phases of the problem. Money had never been available to even tackle the residence smoke. The expenditures for reduction of smoke had been as follows (exclusive of money spent in investigative work by the city and the Bureau of Mines in 1919 and 1920):

1921	1922	1923	1924	1925	1926
$12,500	$10,000	$10,270	$10,000	$10,500	$3,500

In addition, there had been an expenditure of $2500 to $3500 each year by railroads for their own smoke inspectors.

Figuring on a basis of 145,000 population and using the figure $50,000 per million people, he showed that the city should be spending only $7200, while $10,000 had actually been found inadequate. The 1927 budget request called for $16,000. They expected to spend $5000 in reducing residence smoke and hoped to induce the railroads to increase their expenditures to about $5000, making a total expenditure of more than $20,000 for a city of less than 150,000 people.

Writing on the author's emphasis of the importance of supervision of the firemen in small plants, A. G. Christie[16] said that some plan must be evolved that would provide for this constant watching of the fireman and for checking up his performance.

Referring to the statement that at the proper time, depending on economic conditions, the fuel interests and technologists would meet the situation, Professor Christie suggested the following plan for meeting the situation at once:

Since the small plants could not each employ a combustion expert to put their equipment in first-class condition, and to continue its operation in a smokeless manner, it would seem a logical step to have the owners of these plants combine into groups. Each group would then employ a combustion expert to direct the operation of all the plants, to purchase fuel in bulk to be allocated to each plant as needed, and to hire and direct all boiler-room labor. A plan could be worked out with regard to the distribution of this supervisor's time at the various plants and for the sharing of his salary. The employment of such a man, even at a fairly high salary, would save the owners money. He could save money by buying the best heat values in coal at low cost, as he would buy in quantity, and furthermore he would save money through the more efficient operation of the boiler plants. There would be the added social saving to the community due to a reduction of the smoke nuisance, for increased furnace efficiency, he explained, was generally accompanied by decreased smoke.

Theodore B. J. Merkt[17] believed that every member of the Society should do his share in placing the problem before the public, by gathering some of the salient facts concerning the effects of smoke upon our daily lives and presenting these facts in the form of papers or lectures before various public bodies in our towns and cities. Mr. Merkt saw much good resulting from the discussions with the audiences, but the publicity resulting from newspaper reports of such gatherings would be far greater than could be realized.

A national movement, guided by the medical and engineering professions, would do much to safeguard our resources, our wealth, and our health in the proper presentation of the situation before the people, he believed.

A written discussion submitted by Edward J. Kunze[18] also emphasized the fact that in dealing with the smoke problem it was necessary to deal largely with the human factor. Mr. Kunze contended that the best way to arouse the public from its well-known lethargy was to appeal to persons through their original instinct of self-protection.

Usually, he said, all the public thought necessary was the passing of an ordinance in order to abate an evil. As a specific case, he cited the drafting of a smoke ordinance twenty years ago for the city of Newark, N. J. It contained definite statements of what were violations, based upon the Ringelmann Chart.

The ordinance passed the City Council after much opposition by factory owners, then the citizenry calmly went to sleep—they had an ordinance. The politicians, however, did not sleep.

The conditions one might expect to result naturally followed. The patronage of the job had to go to a certain ward, regardless of the type of man available. The public finally awoke to inquire into the inactivity, to be told through the press that the inspector

[15] Lecturer, Dept. of Mechanical Engineering, University of Utah, Salt Lake City. Jun. A.S.M.E.

[16] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

[17] Industrial Engineer, Brooklyn Union Gas Co., Brooklyn, N. Y. Jun. A.S.M.E.

[18] Associate Editor, *Power Plant Engineering*, Chicago, Ill. Mem. A.S.M.E.

had been illegally appointed. The politicians objected to the technical features of the ordinance and finally had them eliminated, thereby removing the technical qualifications of the inspector.

Ordinances were necessary, the writer pointed out, but the real need, as he saw it, was for a group of public-spirited minute men who were willing to keep on the job every minute during critical periods, and who would maintain an occasional watch over operations even when they appeared to be functioning smoothly.

Mr. Kunze also mentioned the tendency on the parts of many well-meaning citizens to advocate drastic legislation or impractical methods of solution. The decision of what should constitute a proper ordinance, and the proper manner of administering a smoke-abatement department, he pointed out, should rest largely with engineers. A campaign was much weakened when men or women, incompetent in these matters, were called in conference regarding them. Such persons might, however, do much good in establishing civic consciousness, he added, and he urged that their efforts be tactfully directed toward this important phase of the work.

Mention was also made of the tendency of the average person to oppose the intrusion into his affairs of an outsider. The appointment, therefore, of a smoke inspector who might bring him into court always resulted in opposition until the fairness and helpfulness of the inspector had been established. If, however, a commission consisting of three or five prominent citizens, most of whom were engineers, were appointed to act as a Board of Appeals, the objection to what might seem arbitrary rules might be broken down. In this case an offender, who might consider that he has been unjustly dealt with by the smoke inspector, might upon the payment of a fee of, say, $25, bring his case for review. Resort to review by a group of one's peers would not frequently be made, he pointed out, and in time the need for the Board might cease.

He did not advise court procedure until all other means had been exhausted, but when necessary he favored action with graded penalties, in which subsequent offences would be penalized heavier and heavier, including the authority to draw the fires.

The educational campaign, he believed, could best be carried on by collecting, in the office of the smoke inspector, records of the results of changes, and in issuing authentic statements of facts concerning costs of repairing damage done by smoke. The cost of cleaning the outside of a single large office building was amazing, he pointed out, and such information presented by a disinterested person would carry much weight in an abatement campaign.

Definite statements by the smoke inspector as to the equipment that should be installed in any specific instance was not recommended by the writer, since the human factor played a large part in the success of any installation, whether of up-to-date or antiquated equipment.

Surveys, although costly, should be the initial point of attack, for without them there was no basis for comparison and persons might think nothing had been accomplished because smoky conditions had not been greatly changed due to the establishing during the period of enforcement of, for example, many new industries. Further, without a survey, an exact basis for knowing the proportional amount of smoke made by the several classes of smokers in the city would not be available.

Mr. Monnett, in closing, said that the fixing of $50,000 a year per million population as a minimum figure with which successful smoke-abatement work could be carried on, had been just an attempt to set some kind of a mark which would give an idea of the importance of such work. The trouble had been in the past that the size and character of the job had not been realized. It had been considered as a petty city hall departmental job, which could be taken care of by passing some kind of an ordinance. The fact was that it was a job of a magnitude comparable with any other major city problem, such as water supply, sanitation, etc. When this was realized, more progress would be made.

The city of Chicago, with a population of 3,000,000, had for many years been getting along with less than $50,000 a year; this, of course, had been totally inadequate really to cover the ground. When it was considered how pitifully small the appropriations were for this work, in the average city, it was not difficult to see why this problem of smoke had been so slow of solution.

Regarding the suggestion, that small fuel users could pool their interests and thus take advantage of expert engineering advice in solving their problems, it might be of interest to recall that in the city of Hamburg there had been for many years such a plan in existence. This had worked out to great advantage at small cost and was an idea which could be used in similar situations in other cities. As a matter of fact, such a plan had been just ready to be put into force before the war in the Manchester District in England, but this work had been stopped when hostilities commenced; however, during the past season the question had again been taken up with the idea of working out something along this line. It would not be necessary to confine such a plan to high-pressure power plants, but the work could be extended to heating plants of all descriptions, with very great advantage, because from this type of plant we got a large proportion of our smoke.

While the smoke problem was primarily an engineering one, still there were so many factors entering into it other than engineering, that it was rather complex in its aspects. Publicity was an important agent in bringing out the various sides of the question. Efforts along lines dealing with present furnaces and types of equipment should not be relaxed, as it was through these efforts that the way would be paved for the future developments in fuel engineering, which we all saw coming, but which must be prepared for by educational and preliminary work.

Miscellaneous Smoke Problems

In addition to the written discussion, interesting and valuable information was submitted in the form of extracts of papers published in *The Forge*, the official bulletin of the St. Louis Section of the Society. The papers from which quotations were taken were Harmful Effects of Smoke on Health, by A. S. Langsdorf,[19] and Smoke and Plant Mortality, by Dr. George T. Moore.[20] Mr. Langsdorf brought out the fact that it was natural to consider that the incidence of diseases of the respiratory tract would provide one measure of the harmful effects of smoke; and that the prevalence of sinus infections in large cities would furnish another, but, he added, it must be remembered that all of these diseases were the result of bacterial invasions whose propagation and development were conditioned by some or all of the following factors: The congestion of population in the community (or in part of it), which affected the degree of exposure to infection; the vitality of the population, determined in part by its average age and in part by the nourishing quality of the food it consumed, hence related to the economic status of the people concerned; the temperature and relative humidity of the air outdoors and in buildings, determined in the latter by the ventilation facilities in stores, factories and dwellings. In view of these considerations, he showed, it was clear that it was necessary to proceed with caution in assigning to smoke alone a definite degree of responsibility for morbidity.

Continuing, the author said that statistics showing the death rate due to non-tubercular diseases of the respiratory tract, for numerous cities and towns in the United States, Great Britain, and Germany, showed that on the whole there was a correlation between the incidence of such diseases and smoky atmospheric conditions. The parallelism was not absolutely complete, however, for in a few cases cities which were fairly free from smoke showed a disproportionately large death rate due to pneumonia.

To draw absolute conclusions concerning the effect of smoke per se would, he showed, require that observations be made on separate communities which were identical in all respects except that smoke was present in some and absent in others: or else that the observations be made on a single community, successively under clear and smoky conditions, all other conditions remaining the same. But different communities might be radically unlike, even though outwardly similar; thus, it would not always be conclusive to compare statistics of cities in these widely differing communities.

Continuing, he said that the soot particles in smoke-laden air differed from other forms of dust in that they had a definitely bactericidal effect, probably due to the fact that soot carried a considerable amount of phenol. But it was not permissible to conclude that because soot was generally sterile it might be breathed with impunity, for the same phenol which killed the bacteria would

[19] Director, Department of Industrial Engineering and Research, Washington University, St. Louis, Mo.

[20] Director, Missouri Botanical Garden, St. Louis, Mo.

likewise act as an irritant to the mucous membrane of the nose, throat, and bronchial passages. A similar irritation might result from the sulphur dioxide absorbed by the soot, and the effect of the irritation might be to produce an inflammation favorable to the development of bacteria always present in the upper respiratory passages.

Further, according to the author, smoky air, by cutting off the direct sunlight, limited the bactericidal effect of the ultra-violet rays, thus indirectly affecting health.

In the second contribution, that by Dr. Moore, attention was called to the fact that the normal growth of a plant depended chiefly upon light, and the nature of the soil, plus the free exchange of gases from a clean leaf surface. According to the author, smoke seriously affected all of these factors, consequently the vegetation of a specific area afforded the most tangible evidence of the presence of the various deleterious substances discharged into the air because of incomplete combustion.

If smoke were nothing but finely divided particles of pure carbon, plants would suffer but little from it, for even if it covered the leaves it would be innocuous and wash off with the first rain. But the tar, practically always present, formed an insoluble layer. More serious than either of these substances, however, were the small quantities of sulphur, arsenic, or carbolic-acid-like products in the smoke, any or all of which might occur when coal was imperfectly burned.

One had but to visit a smoke-infested region to be impressed by the scarcity if not entire absence of flowers on some of the most beautiful shrubs, he said, adding that not only the plants out of doors but those under glass suffered seriously from the toxic substances found in the air. Citing a specific case, Dr. Moore mentioned the effects of a smoke cloud in St. Louis during November a few years ago, which so affected many of the flowering plants that the leaves dropped over night and the foliage of others was so discolored and burned that only a small percentage survived. The plants appeared to have been sprayed with a strong solution of sulphuric acid. In the words of Dr. Moore, "If all vegetation suffers as it obviously does from smoke, what is happening to human beings?"

Victor J. Azbe,[21] also reading from *The Forge* and from a paper prepared by himself on Development of an Efficient and Smokeless Firing Method for Domestic Furnaces, described a system of baffles developed for this type of furnace which supplied sufficient air at the right point to insure complete combustion of the combustible gases released in the furnace. A down-draft baffle permitted large quantities of coal to be fired at one time. Coal back of this baffle being protected from the high heat in the center of the furnace, the volatile matter was forced to pass through the incandescent coal, where there was a tendency to break the hydrocarbons into fixed gases so that they burned with much greater rapidity. A second baffle above the first acted as a reflector of heat to the combustion zone, further lengthening the path of travel of the gases, which would short-circuit to the stack if the baffle were not present. Some of the advantages claimed were: No smoke at any time; greater furnace capacity possible when needed; no carbon monoxide; high CO_2; better burned-out ash; longer between firing periods; better results with lower grade fuel; firing simpler; applicable to most existing installations.

Contributing to the oral discussion, Warren Viessman[22] mentioned the rather great effect of smoke on property and on vegetation. The reported costs per capita in several cities, he said, were twelve dollars in Cleveland, eighteen dollars in St. Louis, and ten dollars in Baltimore. These figures, which he felt to be low, included cost of cleaning buildings and damage to structures and paint, as well as increasing laundry bills.

The domestic problem of smoke, he felt, was probably of more interest in some communities than in others. In Baltimore, for instance, the residents in the past had used hard coal in their home furnaces, but the rise in the price of hard coal in that part of the country had rapidly increased to the point where the people were adopting other fuels, including soft coal, which they did not know how to fire; many of the residents attempted to use the soft coal in the same manner as the hard coal, consequently there was a great deal of smoke. Adoption of the type of furnace outlined in Pamphlet 273 of the Bureau of Mines would improve conditions considerably in apartment houses and residential districts, he felt.

Attempts had been made to arouse some interest in the smoke abatement ordinance in Baltimore, he said, but the public had not yet taken to it; they had not been properly educated to it. He felt that it was the duty of the engineers in each of the communities to shed as much information as possible on the subject, to arouse the public to the requirements of the smoke ordinance and the benefits to be gained from it, and also to impress upon the plant owners the need for economy in the burning of fuel, and the resultant smoke elimination through that method would also be of importance.

McRea Parker,[23] describing conditions in Cleveland, brought out the desirability of placing a man of experience in charge of a project such as smoke abatement and then seeing to it that he had the necessary coöperation. In his opinion, one of the greatest forces in the successful operation of such a program was the assumption of the attitude of a public servant, rather than one whose business it was to enforce a law. In many cases the inspectors were prepared to make tests where the plant was not properly equipped with instruments.

He favored the plan suggested by Professor Christie, namely, the employment of an expert by a group of plants to study their several problems, and said that such a plan was being carried out by two organizations of business men in Cleveland, representing certain classes of building owners.

Elliott H. Whitlock,[24] also speaking of conditions in Cleveland, said that they were often met by a request to clean their own buildings before approaching other offenders. Therefore, they first checked up on the fire houses and police stations, investigating the coal problem and making recommendations as to the best coals to use and the most desirable methods of firing. The result was that they were soon approached by others desiring surveys of their buildings. Results of these investigations for others had proved very satisfactory and the savings to the owners and improvements in cleanliness had made friends where friends were most desirable.

Dr. Bato referred briefly to some of the furnaces and stoves designed for burning the brown coals of Germany. In these stoves the fuel magazine was so placed that the flames entirely surrounded it, which insured complete drying and coking of the fuel before it reached the grate.

Morgan B. Smith,[25] discussing the Detroit ordinances and attempts to prevent smoke, said that their chief difficulty with their old ordinance had been to obtain convictions. Cases were thrown out of court, simply because there was no way of defining dense smoke. In cases where the Ringelmann smoke chart had been used, the judge had always taken the stand that no man could check himself, nor could any two men check each other. He advocated the use of some means that would determine the density beyond reasonable doubt.

Electricity Used for Melting Metals

THE melting of certain metals by the use of electricity as compared with coal-fired furnaces has been described by the Chief of the Metallurgical Division of the Bureau of Standards in a statement recently released for publication.

The paper points out that aside from electric reduction in fused salts and the making of ferroalloys, electrothermic reduction of ores of the non-ferrous metals is not commercially practiced in America. Electrothermics finds some application in annealing and heat-treatment of non-ferrous alloys, but its chief application is in melting. About 25 per cent of the type metal is electrically melted. Electric melting is practically standard for nickel and its alloys. Around 90 per cent of the output of brass, bronze, and nickel silver from the rolling mills, and nearly 30 per cent of the output from foundries, is melted in electric furnaces.

[21] Consulting Engineer, St. Louis, Mo. Mem. A.S.M.E.

[22] Industrial Power Engineer, Consolidated Gas, Electric Light and Power Co., Baltimore, Md. Jun. A.S.M.E.

[23] Mechanical and Electrical Engineer, Cleveland Worsted Mills Co., Cleveland, Ohio. Assoc-Mem. A.S.M.E.

[24] Commissioner, Smoke Inspection, City of Cleveland, Ohio. Mem. A.S.M.E.

[25] Engineer, General Motors Corporation, Detroit, Mich. Mem. A.S.M.E.

The Lubrication of Waste-Packed Bearings

By G. B. KARELITZ,[1] EAST PITTSBURGH, PA.

In this paper, which is based on results of an investigation made by the Westinghouse Electric and Manufacturing Company, the author discusses the feeding of oil through waste and the existence of a load-carrying oil film in waste-packed bearings as essential for their proper performance. It gives observations on the friction and temperatures of these bearings and on the importance of proper packing to insure sealing the window by oil-saturated waste, and also discusses the existence of a critical oil lift at which the seal is broken. The reason for occasional end wear and scoring of the ends of a shell during the running-in period of service is given.

THE small space available in a car truck excludes the possibility, with the present arrangement of motors and gears, of using a bearing well sealed against end leakage with "perfect" lubrication. Moreover, the constant increase in the power, size, and speed of railway motors has led to more severe conditions in the bearings. Unless radical changes are made, such as the application of roller bearings, or of double universal-joint propelling shafts,[2] etc., the use of waste-packed bearings of the sleeve type will have to be continued. So far nothing has been found that would replace these waste-packed bearings, which are simple and rugged, in spite of the dependence of their performance on proper packing.

Technical literature contains surprisingly few data on the basic principles of operation of such bearings, owing to the expense and difficulty of experimentation. The following notes on the performance of waste-packed bearings, the result of an investigation undertaken by the Westinghouse Electric and Manufacturing Co. may therefore be of interest.

Experiments were made on several 5-in. by 9-in. split bronze axle bearings. Although the loads and speeds on these bearings were much lower than those encountered on armature journals, the general principles of operation were brought out distinctly and most of the findings may be applied to any type of waste-packed bearings.

Oil-Lifting Capacity of Waste

A waste-packed bearing consists of a shell, solid or split, which is pressed or clamped tightly in the surrounding housing, with a window into which waste is packed. Oil is supplied to the bearing by the waste which communicates with an oil reservoir and lifts it to the journal by capillary action. For satisfactory performance a bearing requires a certain minimum supply of oil and a window reasonably well sealed by the oil-saturated waste.

The amount of oil carried through a wick and the height to which it may be lifted depend on the kind of waste and oil[3] used and on the oil temperature. When a long wick made of several strands of a given waste material is dipped into oil the capillary action causes the oil to rise to a certain height. This depends on the surface tension of the oil and its adhesion to the material of the wick. The lift is an increasing function of surface tension and adhesion. Another factor influencing the height of lift is the size of the microscopic channels in the waste through which the oil flows. The smaller these are, the higher is the lift.

Fig. 1 gives the height of lift[4] for a light air-compressor oil (specific gravity, 0.872; viscosity, 53 sec. at 100 deg. cent.) and for a heavy cylinder oil (specific gravity, 0.905; viscosity, 630 sec. at 70 deg. cent.) with different materials.

When a dry wick is dipped into oil the height of lift changes rapidly at first, then the rate of rise decreases, the lift finally attaining a constant value after a day or two. Fig. 2, plotted from test results obtained by the author, shows the height of lift against time for wool waste with three oils at a room temperature of approximately 30 deg. cent.

Oil	Description	Viscosity, sec. 100 deg. fahr.	130 deg. fahr.	210 deg. fahr.
A	Medium machine oil	229	116	52
B	Heavy machine oil	625	273	68
C	Summer car oil	690	307	73

Oils A and B were submitted as overhead distillates of an asphalt-base crude, the sample C as a residue oil of a paraffin-base crude.

The ultimate heights of lift for higher temperatures are slightly lower, since the surface tension of oils decreases slowly with an increase in temperature. This explains the fact that in waste-packed bearings the oil level rises in the oil chamber after the bearing warms up, the difference in the cold and hot oil level being observed to be from $^1/_4$ to $^1/_2$ in.

It has been found that the amount of oil a wick of a given size

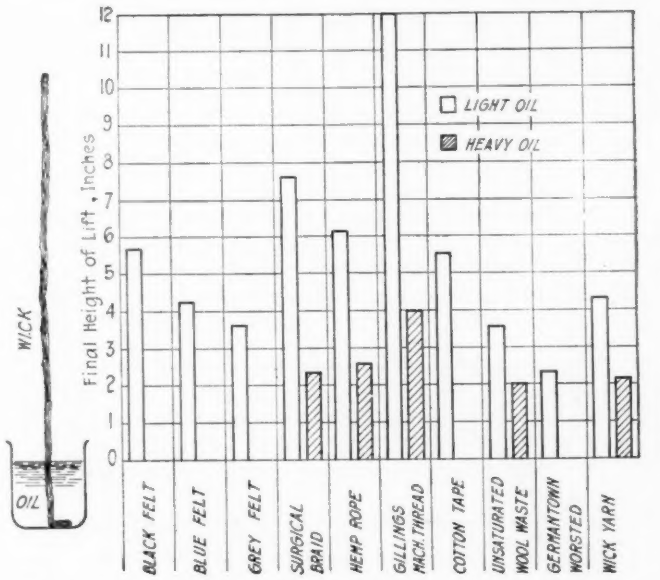

Fig. 1 Height of Lift for a Light Oil in Various Materials

and of different material is able to siphon over is not correlated with the heights given in Fig. 2. The gravity and capillary forces acting upon the oil and causing it to flow, work against the internal friction or viscosity of the oil and against the friction between oil and waste. It is quite natural that in some materials with high lifting capacity this friction is considerable due to the small capillary channels, which will cause a small flow of oil. The commonly used materials when graded for their ability to transfer oil come in the following order: lamp wick, cotton waste, felt, wool waste; the lamp wick being the best and the wool the worst. Cotton materials, so superior in lifting and feeding capacity, are unsuitable as packing for bearings because they glaze readily, and, having no inherent elasticity, the positive contact with the journal is easily destroyed by vibration and jarring of the car on the track. The only material that has been successfully used as packing is wool waste or a mixture of wool and cotton. When, however, the mechanical properties are of secondary importance, as in auxiliary oil drippers, siphon feeds, etc., lamp wick or cotton thread should be used.

At high temperatures, while the surface tension, capillary action and specific gravity change but slightly, the viscosity of oil decreases many times; consequently, the flow of oil increases very rapidly with rise of temperature. Figs. 3 and 4 show the flow of oil through a cotton wick for oils B and C, respectively. The wicks were made of 26 strands of lamp-wick cotton; their weight[5] was 0.182 gr. per in. Although tests had shown no appreciable

[1] Research Department, Westinghouse Elec. & Mfg. Co.

[2] Used in Germany with motors mounted on the car frame.

[3] See Packing Railway Motor Bearings for Oil Lubrication, by C. Bethel, in *Electric Railway Journal*, April 19, 1924.

[4] Data obtained by B. F. James, formerly of Westinghouse Electric & Manufacturing Co.

Presented at the Annual Meeting, New York, December 6 to 9, 1926, of The American Society of Mechanical Engineers. Abridged.

[5] It was found that the amount of oil fed is roughly proportional to the cross-section of wick, all other conditions being equal.

difference in the static heights of lift for B and C through cotton wick and the viscosities of the two oils are quite close to each other, the amounts of oil fed differ very much. Probably this may be explained by the fact that oil C had in it insoluble matter which might clog up the minute channels in the cotton. In both cases the temperature is the most decisive factor influencing the flow of oil.

The above explains the fact that under heavy loads and after an appreciably long period of rest, i.e., when cooled thoroughly, bearings have been noticed to run warmer than usual. This was caused by the fact that no oil was flowing into them until the oil and waste had heated up.

The dependence of the flow on the "oil lift" in a bearing, i.e., on the distance between the oil level and the window, can be appreciated easily from Figs. 3 and 4. Quite naturally the oil flows more freely just after filling the bearing, this flow gradually decreasing in quantity toward the end of the oiling period. This was illustrated by an experiment with a well-run-in axle bearing shown in Fig. 5. The bearing, 5 in. in diameter by 9 in. long, belonged to a 60-hp. railway motor. It was run at 350 r.p.m. with a total load of 1120 lb. This corresponded to service conditions at 32 m.p.h., assuming 33-in. wheels. Oil C was used for lubrication. The bearing was run for a certain time, then stopped and cooled. Fig. 6 shows the relationship between oil lift, oil consumption, and the mileage run after oiling. After 6250 miles the friction became

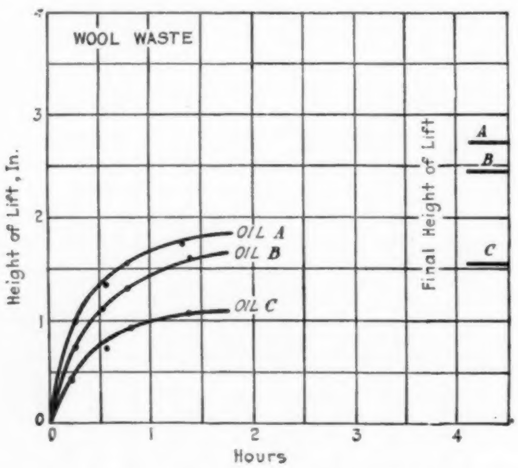

Fig. 2 Height of Lift Against Time for Wool Waste

large and irregular and the temperature rose sharply, indicating that the window had become unsealed and that no adequate oil film was building up. Evidently, further running without reoiling would have injured the bearing.

It will be noticed later that the performance of a waste-packed bearing does not depend on the amount of oil flowing through it, provided the amount of oil supplied to the journal is sufficient to keep the bearing clearance full. Therefore it appears that the high rate of flow during the first 1500 miles after reoiling leads to a waste of oil. Of the 300 grams lost by the bearing during the whole period at least 100 grams could be saved if a constant, adequate flow and constant oil lift were maintained. Nevertheless, filling the oil chamber to a high level while reoiling is necessary, otherwise there is no assurance that the waste, which is usually fairly dry at the end of the oiling period, will be saturated quickly enough to provide a good seal for the window. In the new type of "oil-sealed" motor housings the oil lift is automatically maintained at a predetermined value and service experience has shown a corresponding increase of the oiling periods for these bearings.

The Oil Film in Waste-Packed Bearings

When a journal comes to rest, part of the oil stored in the clearance flows out of the bearing at the ends. A certain part of it is kept inside by the capillary action between the oil, shell, and journal, and collects at the bottom of the bearing. In the case of axle bearings where the shell rests on the journal, the capillary action keeps oil also at the line of contact on the top of the bearing.

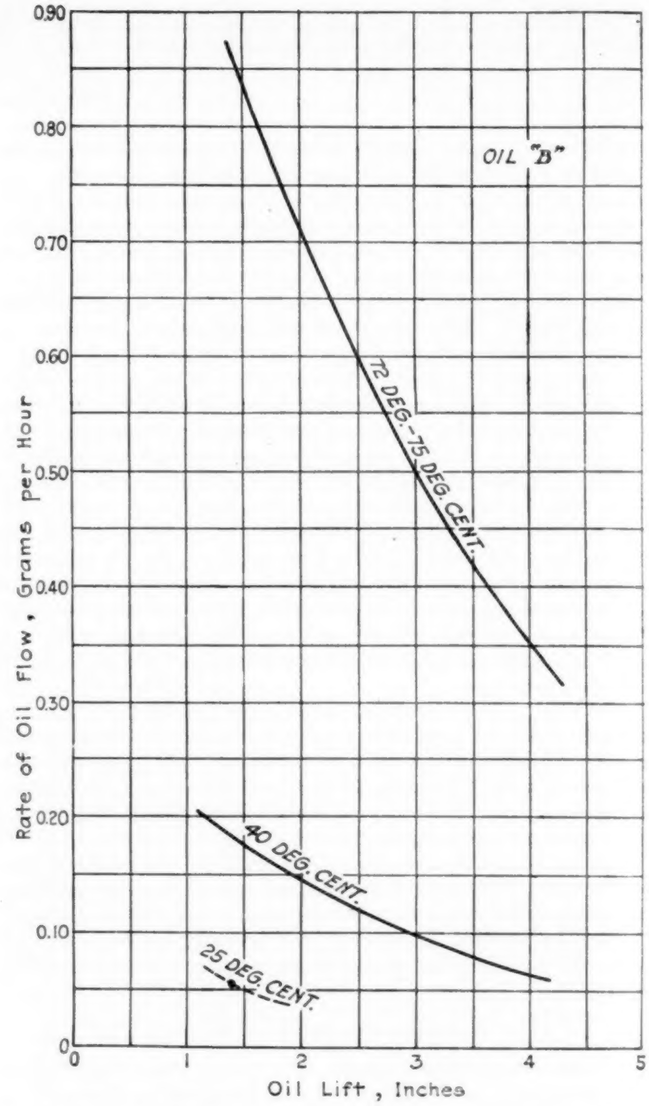

Fig. 3 Flow of Oil Through a Cotton Wick for Oil B

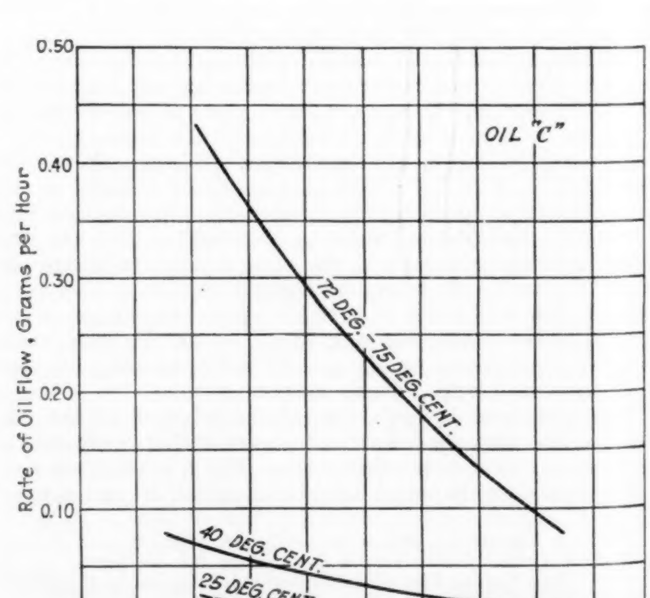

Fig. 4 Flow of Oil Through a Cotton Wick for Oil C

After the journal is started this oil lubricates the bearing even during the first moments of rotation. The starting torque of the journal corresponding to a coefficient of friction of from 12 to 18 per cent, the running friction drops instantly to a value corresponding to a coefficient of 6 to 8 per cent, provided the speed of rotation is not too low. Under proper conditions, in a few revolutions the journal wipes off the waste a sufficient amount of oil

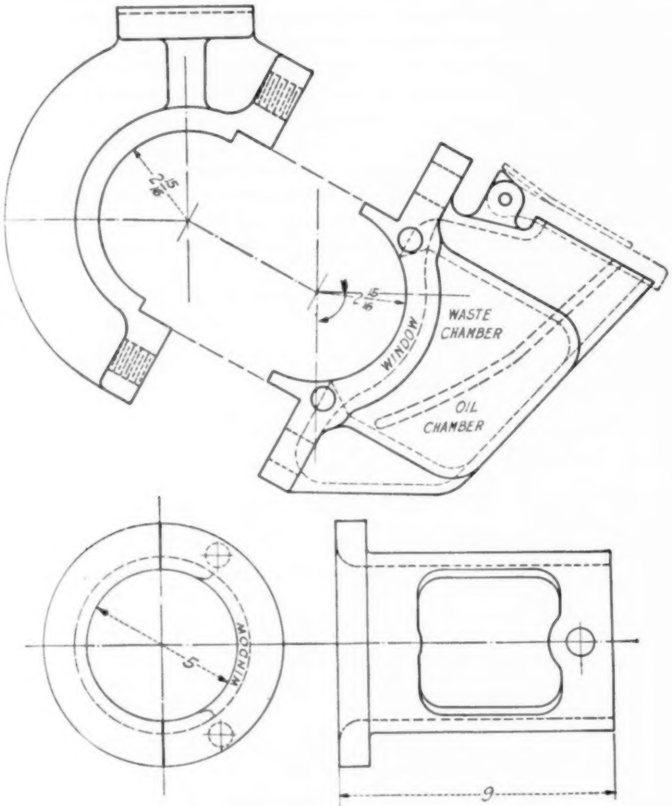

FIG. 5 AXLE BEARING USED IN THE EXPERIMENTS

to fill the whole clearance, while the oil is also gradually squeezed out toward the ends of the bearing.

In our tests the very sensitive recording torsiometer used with the testing machine (see Appendix) showed erratic fluctuations in friction whenever the lubrication of the journal failed over a part of its length. On many occasions, due to some disturbance in the bearing, such as dirt or slush in the clearance, lack of proper lubrication could be observed until the bearing was warmed up sufficiently to produce a more ready flow of oil, reëstablishing the oil film. The friction in the bearings with an incomplete oil film was so high that they could not run for a long time under such conditions without being overheated. A bearing which runs satisfactorily in service must necessarily have a clearance completely filled with oil.

The performance of a waste-packed bearing is therefore subject to a large extent to the same hydrodynamical laws as those governing the operation of "perfectly" lubricated bearings, although the oil films in these two types of bearings have different characteristics.

The action of the oil film in a bearing with perfect lubrication, that is, one running in an oil bath or with an unlimited or large supply of oil, is well described in the literature on the subject.[6] The boundary layers of oil are kept to the surfaces of the shell and journal by adhesion. A relative motion of oil layers is thus created inside the clearance, the velocity gradient across the small clearance, from journal to shell, being very high. Due to the viscosity of the oil, the slipping of oil layers over each other results in forces of two kinds:

a Friction forces resisting the motion of the journal and measurable in terms of torque on the shaft

b Hydrostatic pressure sufficient to carry the load on the journal, separating the journal and shell by a steady oil film. The pressure is built up on the "down" side of the journal (for a downward load), the "up" side having a vacuum (Fig. 7).

In a bearing of finite length the pressure at the ends is necessarily equal to atmospheric pressure, and the hydrostatic pressure must vary in the longitudinal direction as well as along the circumference. The vacuum on the up side is small, as it cannot exceed the vapor pressure of the oil at the bearing temperature. A pressure gradient from the middle of the bearing toward both ends causes an axial flow of oil through the clearance, the bearing acting thus as a pump. "Perfect" lubrication is maintained when the supply of oil to the bearing is not less than the pumping capacity of the bearing under the given conditions of load, oil, temperature, and speed. This is the case with oil-ring, oil-chain, or forced-feed bearings.

On the other hand, the amount of oil supplied by the waste is negligible when compared with the pumping capacity of the bearing. Therefore, assuming that a journal-carrying oil film were established in some way, oil would flow out of the bearing ends until the journal seated itself on the shell. Boundary lubrication[7] would exist along the contact area between journal and bearing. After such a contact is established, the oil flow through the bearing is very small, the oil being kept inside the clearance partly by the capillary action of the clearance and partly by the light vacuum created on the up side.

The shape of the oil film in a waste-packed bearing is determined only by the machined clearance of the bearing. On the other hand, in a perfectly lubricated bearing the oil film adjusts itself to each condition of service. The equilibrium conditions for these bearings are usually such that the maximum pressure of the oil is about three

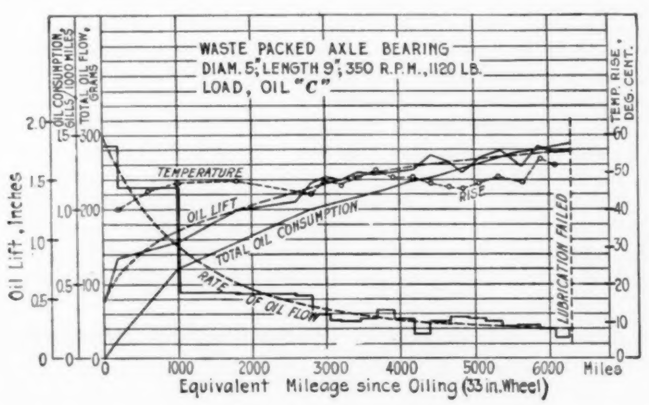

FIG. 6 RELATION BETWEEN OIL LIFT, OIL CONSUMPTION, AND MILEAGE

times the average pressure on the projected area of the bearing. It is known that this pressure depends on the eccentricity of the journal in respect to the shell, increasing very rapidly when the eccentricity nears half of the clearance, that is, when the journal nears contact with the shell. Therefore the pressures in oil films of waste-packed bearings may be found greatly exceeding those in perfectly lubricated bearings. In the 5-in. by 9-in. axle bearings tested by the author a total load of 1200 lb. at 450 r.p.m. gave a pressure of 900 to 1000 lb. per sq. in. on a gage located at the mid-length of the bearing and 30 deg. behind the load line with respect to the direction of rotation. The nominal pressure with reference to the projected area was only 30 lb. per sq. in. (considering the bearing to have a length of 8 in. due to the radius of the bearing collar).[8]

In contrast with the perfectly lubricated bearing, the load in a waste-packed bearing is now shown to be carried partly by hy-

[6] H. A. S. Howarth, A Graphical Study of Journal Lubrication, Part I to III. Trans. A.S.M.E., 1923–1925.

G. B. Karelitz, Charts for Studying the Oil Film in Bearings. Trans. A.S.M.E., 1925.

L. Guembel—E. Everling, Reibung und Schmierung im Maschinenbau (Friction and Lubrication of Machinery), Berlin, M. Krayn, 1925.

[7] A clear picture of this type of lubrication, when the oil film is too thin for hydraulic phenomena to take place, has not been given to date.

[8] Compare with the article of Erich Schulze, Studien über Achslager für Fahrzeuge (A Study of Car-Axle Bearings), in *Verkehrstechnik*, June 25, 1926.

draulic pressure and partly by the contact area. The percentage of the total load carried by each can be estimated by considering the friction. Let P_1 represent the part of load carried by the oil film and P_2 the part carried by the contact area; then $P = P_1 + P_2 =$ total load on bearing.

The coefficient of friction in the oil film may be taken equal to 0.01, corresponding to the high-viscosity oil used in waste-packed bearings. The coefficient of friction in the contact area is in the neighborhood of 0.12 for conditions of boundary lubrication. The overall coefficient of friction has been observed to be of the order of 0.03. Then

$$0.01P_1 + 0.12P_2 = 0.03P = 0.03(P_1 + P_2)$$

or

$$0.09P_2 = 0.02P_1; \qquad P_1 = 4.5P_2$$

Although the oil film carries most of the load, the contact area creates most of the friction:

Friction in oil film is $0.01P_1 = 0.045P_2$
Friction in contact area is $0.12P_2$

The above shows that in waste-packed bearings the viscosity of the lubricant is as important as its oiliness. Since these bearings run normally at a temperature near 80 deg. cent., a high viscosity at room temperature is essential.

The load-carrying ability of the oil film is so important that the design and manufacture of bearings should be such as not to interfere under any conditions with its being formed and remaining uninterrupted.

The window in a bearing must be located along the circumference in such a way as to provide an ample angle between the window and the load line under any conditions of service, in order that an efficient load-carrying oil film may be established.

For instance, the test bearings, when loaded as shown in Fig. 5 and run in the indicated direction, showed at different loads and speeds a friction torque of from 1.4 to 2.6 times lower than the torque for the opposite direction. (An analysis of the forces acting in a railway-motor and gear system shows that axle bearings when rotating in the unfavorable direction are usually loaded but lightly. If heavy loads should occur while running in this direction, high temperatures should be expected.)

The usual construction of railway motors is such that under all possible conditions of service the window is in the vacuum zone of the oil film. This helps the waste to feed oil to the journal. But the packing must be made carefully in order not to impair the vacuum which is partly responsible for keeping the oil inside the clearance. The packing must cover the whole window so that it is completely sealed by the oil-saturated waste. The warmed up. No friction change occurred after a steady temperature of the bearing was reached. Fig. 8 shows a series of friction records taken on the experimental axle bearing under 1200 lb. load at 350 r.p.m. with different oil levels. The bearing was well run in. The average growth of the temperature with time is also plotted in the figure.

It will be noticed that a coefficient of friction of 0.01 corresponds to a friction torque $0.01 \times 1200 \times 2.5 = 30$ in-lb. The coefficient of friction therefore varies from 0.075 when the bearing is cold to approximately 0.0175 when the bearing is hot.

Since the coefficient of friction under boundary conditions of lubrication due to oiliness of the lubricant has been found to vary slightly with change of temperature,[10] the variation in friction as found is therefore to be ascribed to the decrease in viscosity of the oil with temperature rise. A typical relation between friction and viscosity is given in Fig. 9. The curve was derived from the known relation between the viscosity and temperature for oil C used in this experiment, and from the observed friction-temperature function.

This change in viscosity of the oil no doubt affects the load-

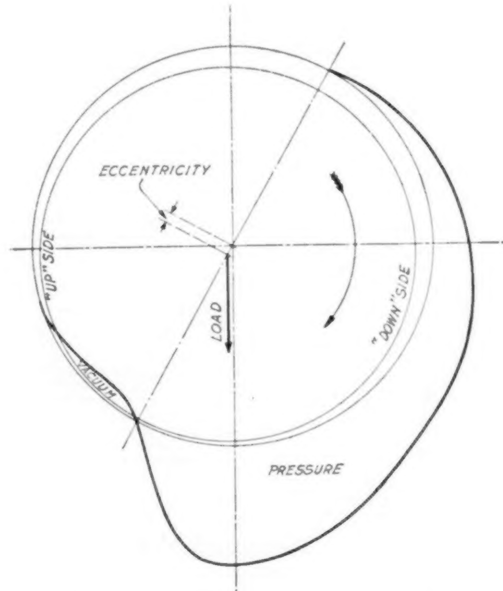

Fig. 7 Hydrostatic Pressure in a Bearing with Perfect Lubrication

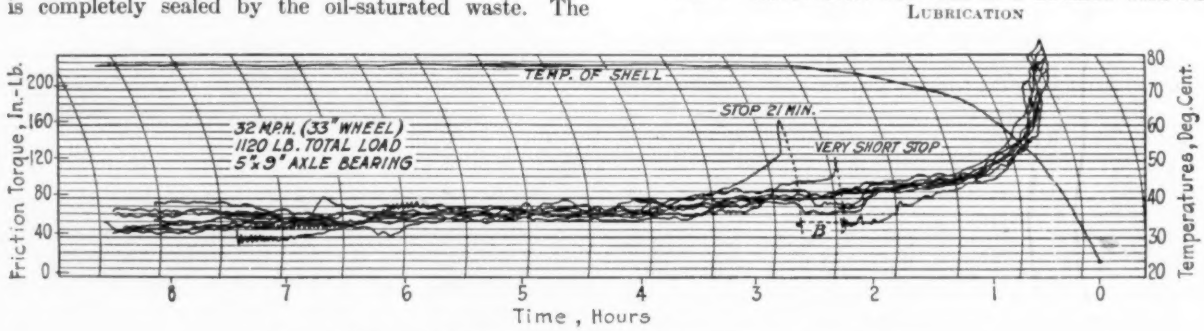

Fig. 8 Friction Records Taken from Experimental Axle Bearing

easiest way to pack with this in view is by placing a wick long enough to reach the bottom of the oil well, across the window and filling the waste chamber behind the wick with waste. This must be tamped very tight so as to force the wick against the journal and to prevent any loosening of the waste in service due to vibration and jarring.[9]

Friction and Temperatures

Influence of Temperature on Friction. In our tests it was observed that after starting the journal, under various conditions of load and speed the friction decreased considerably as the bearing

[9] See article by C. Bethel in *Electric Railway Journal*, April 19, 1924.

carrying capacity of the oil film so that the distribution of the load and friction between film and contact area varies with temperature. An increase in viscosity relieves the contact area, and the distribution of friction should be of a character as represented by the dotted line in Fig. 9.

Influence of Oil Lift. The amount of oil the waste can supply is not sufficient at the best to produce a change in the friction, being too far below that necessary to bring about a state of perfect lubrication. The performance of the bearing is practically independent of the oil flow, provided the clearance is full of oil. This can be seen from Fig. 6 where temperatures are given for different

[10] Mésure de Frottement Onctueux (Measurement of Greasy Friction) by Paul Woog, *Comptes Rendus*, vol. 180, pp. 1824–1826.

oil lifts after steady conditions were attained. It will be noted that the bearing ran distinctly warmer only during the last 200 miles of the oiling period, when conditions grew unsteady and the clearance was not completely filled with oil.

Temperatures at Different Loads and Speeds. A series of experiments was made with varying loads and speeds until steady conditions of temperature and friction were attained. The temperature rise of the bearing is given in Figs. 10 and 11. These show that speed has a considerably more pronounced influence on the temperature than the load. The final temperature attained is determined by the equilibrium of heat generated in the bearing and dissipated through radiation.

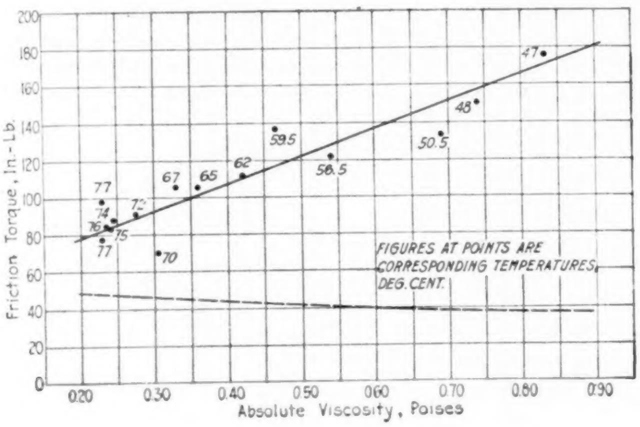

FIG. 9 RELATION BETWEEN FRICTION AND VISCOSITY

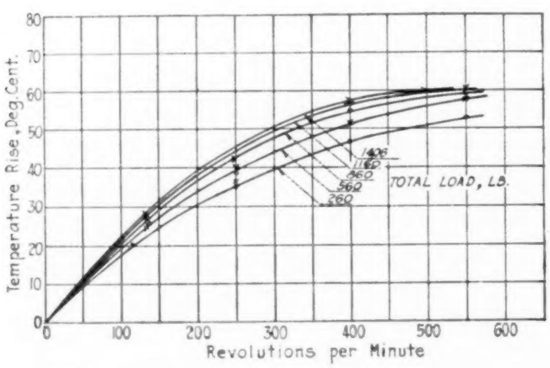

FIG. 10 TEMPERATURE RISE IN BEARING

The generated heat is proportional to the work of the friction forces:

$$H_g = K \frac{T}{r} V$$

where H_g = rate of heat generation
K = coefficient of proportionality
T = friction torque on journal
r = radius of the journal
V = peripheral speed.

The dissipated heat may be represented by $H_d = C(\Delta t)^m$, where H_d is the rate of heat dissipation, m and C are constants for a given bearing, and Δt is the temperature rise of the bearing. At steady conditions

$$K \frac{T}{r} V = C(\Delta t)^m$$

or

$$(\Delta t)^m = \frac{K}{Cr} TV$$

The shape of the oil film being constant in a waste-packed bearing (the temperature distortion not being taken into consideration), the liquid friction is proportional to the speed, and the torque T is an increasing function of both load and speed. The temperature rise being a function of the product TV, the stronger influence of the speed as compared with that of the load is evident. The relation between temperature, load, and speed is complicated by the fact that the torque T, depending greatly on the viscosity of the oil in the oil film, is a decreasing function of the temperature of the bearing. This[11] accounts for the flat shape of the curves in Figs. 10 and 11.

Limiting Pressures and Speeds. A formula which would give the interrelation between all factors in a waste-packed bearing would be very complicated and its use would require a preliminary determination of many constants dependent on the bearing construction. Simple formulas used frequently (for instance, the formula for limiting loads and speeds, $PV = C$, where P and V are pressure per square inch of projected area and the peripheral velocity, respectively, and C is a constant) can serve only for expressing in an approximate way results of experience, covering a limited range of conditions. It is the author's belief that tables and charts representing data collected on the performance of bearings are of more value for guiding designers than such formulas. Charts show the

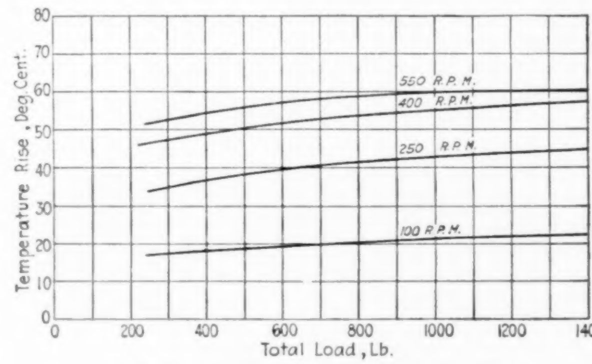

FIG. 11 TEMPERATURE RISE IN BEARING

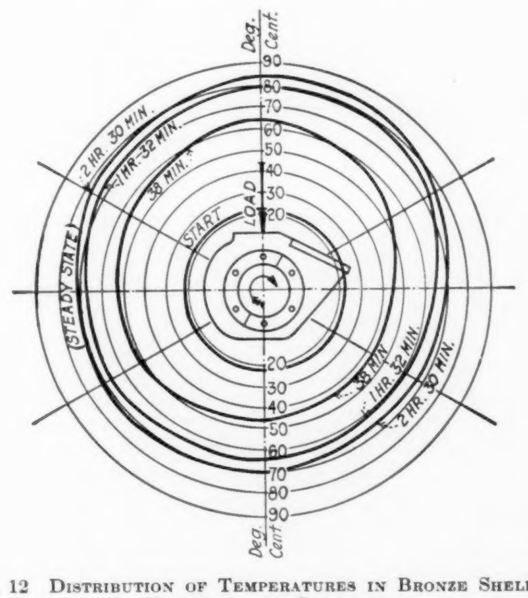

FIG. 12 DISTRIBUTION OF TEMPERATURES IN BRONZE SHELL OF EXPERIMENTAL BEARING

range of values from which they are constructed, and this eliminates the danger of extrapolating beyond legitimate limits.

Distribution of Temperatures in the Bearing. Since the heat is generated in a small region near the line of contact and since the shell is tightly fitted into the housing, the temperature of the bearing varies considerably at different points. Fig. 12 shows the distribution of temperatures in the bronze shell of the experimental bearings, as measured by thermocouples imbedded into the shell at the middle. The difference between the top and bottom for

[11] It should be noted that the pressures and velocities in these experiments were quite low when compared with those met with in armature bearings of railway motors. The shape of similar curves for higher speeds and pressures might be different, and further tests are in progress.

steady conditions was 14 deg. cent., and this difference was still higher shortly after the bearing was started. There was also a variation of temperature along the journal of 2 to 5 deg. when the conditions were stable. When the oil film was disturbed for some reason, a sharp rise of temperature at one end or the other was observed following an increase in friction recorded by the torsiometer. No doubt this was due to interruption of the oil film at that end. After restitution of the lubrication the temperature would quickly even out along the journal.

Temperatures at the center line of the shaft at midlength of the bearing were recorded by means of a thermometer inserted into an axial hole drilled into the shaft, and rotating with the journal. It invariably registered a temperature approximately 4 deg. cent. higher than the temperature of the shell at the hottest spot. It can be assumed, therefore, that temperature of a journal is the same as the average temperature of the oil film.

Critical Oil Level

In order to demonstrate the absolute importance of maintaining a good oil film throughout the whole bearing, experiments were made repeatedly to observe the influence of lowering the oil level in the oil chamber.

It was found that after reaching a certain oil lift ($1^1/_2$ to $1^3/_4$ in. for this bearing), the load-carrying oil film was ruptured. This rupture can be explained on the basis that the capillary action of the waste was not sufficient to hold at such an oil lift enough oil to seal the window and to supply to the bearing the very small amount of oil necessary to replace the loss through end leakage or evaporation, thereby allowing air to enter the vacuum zone of the clearance.

During a typical experiment of this kind, by pumping the oil

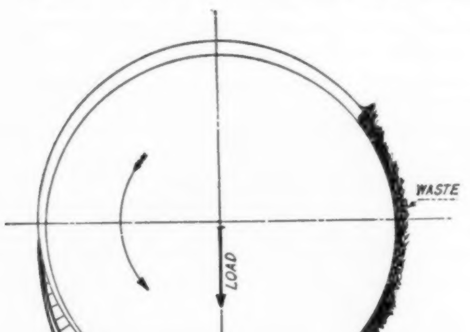

Fig. 13 Effect of Oil Grooves on Oil Film

out of the oil well in small quantities the oil lift was gradually increased. When it reached $1^9/_{16}$ in. a rapid increase in friction occurred. The erratic friction record indicates a ruptured oil film. At this point the friction torque reached 120 in-lb., as compared with 46 in-lb. shown before the disturbance. The temperature curves showed that the oil film was ruptured at one end of the bearing. A further increase in oil lift, to $1^3/_4$ in., caused another sudden change in friction torque, bringing its value to 200 in-lb. The temperature record shows that this was due to the additional rupture of the film at the other end of the bearing. The oil well was then refilled with oil to an oil lift of $^7/_8$ in., but the friction did not change. A vigorous "poking" of the waste with a packing iron reëstablished the oil film and the friction dropped suddenly to the original value.

Grooving of Bearings and Finish of Surfaces

The design of grooves in a waste-packed bearing must take into consideration the proper formation of an uninterrupted oil film. The location of grooves in a pressure zone is detrimental to the performance of a bearing, which is illustrated in Fig. 13. The load-carrying oil film, being broken into two parts, loses part of its carrying capacity, thus throwing more load on the contact area and increasing the friction considerably.

Grooves must be well chamfered or rounded, otherwise the sharp corner will disrupt the continuity of the oil film, and sharp edges on the grooves may scrape the oil from the journal. With the bearing at rest and well cooled, no oil could be found in the grooves after carefully dismantling the bearing, the capillary action of the clearance being sufficient to draw the oil out of the grooves. They cannot therefore serve as reservoirs for oil in the bearing while it is at rest.

In general, the usefulness of grooves in waste-packed bearings for oil transmission along the bearing is doubtful, and a properly designed waste-packed bearing will function just as satisfactorily without grooves.

Regarding the nature of the bearing surface, there is very little oil in the clearance after the bearing is started, and it takes a certain time until a complete oil film is built up (see the friction record on

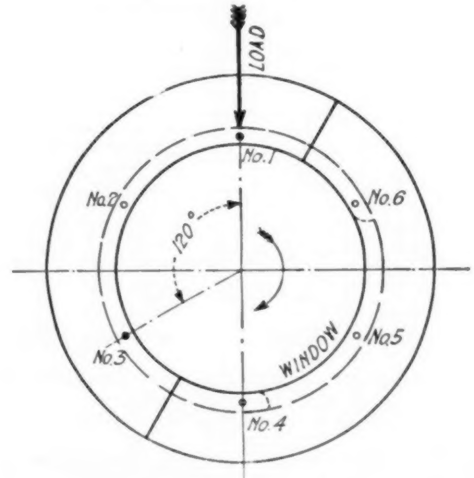

Fig. 14 Locations of Thermocouples in Bearing Shell

Fig. 8 where a very short stop was made). It is therefore very important to finish the surfaces of the journal and shell as well as possible.

Running-In Period of Service

As a rule, new waste-packed bearings, while being run in, wore from the ends to the middle. After several hours of running it was found repeatedly that a shell was scored at the ends, yet after cleaning and scraping, the bearings gave satisfactory service. An investigation of this phenomenon showed it to be the result of a distortion due to uneven heating of the shell. The bearing bends in such a manner as to prevent the middle part of the shell from coming into contact with the journal. The journal rides on both ends of the shell, provided there is no misalignment between the bearing and shaft. Experiments were carried out to determine the cause of such a distortion and to measure the magnitude of it. The 5-in. by 9-in. axle bearing was used. Angle bars were clamped tightly to a rib of the housing and the collar of the shell, and a mirror extensometer was used to measure the change in distance between the top ends of these angle bars. The knife edges of the extensometer were held between the top of one bar and a spring strap bolted to the second bar, so that any relative movement in the angle bars resulted in a minute turning of the mirror observed by means of a telescope and scale. Thermocouples were inserted into the shell of the bearing in the positions shown in Fig. 14.

Loading and unloading the bearing at rest and while running failed to produce any appreciable change in the reading of the extensometer. The change in scale reading was plotted against the time from the starting of the bearing. The change was corrected for the comparatively small elongation of the spring strap due to its slight heating. The temperature rise of the bearing with respect to the ambient air and the difference between temperatures registered by the thermocouples No. 1 and No. 3 were plotted. The perfect correlation between the deflection of the bearing and this difference shows clearly that the shell bends under the influence

of an uneven heating of the top and bottom of the bearing (Fig. 15). A simple estimate of the amount of bending to be expected for a certain temperature difference, the coefficient of linear expansion for the material being known, gave results checking closely with the observed values. It was found that the deflection-clearance for the middle-sized bearing used was of the order of 0.0013 in. at the moment of maximum distortion, and 0.0005 in. corresponded to steady conditions with a temperature difference of top and bottom of approximately 13 deg. cent.

Such a temperature variation being inherent for the operation

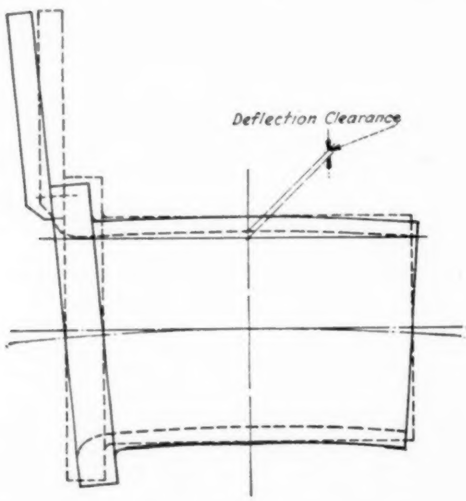

FIG. 15 DEFLECTION DUE TO UNEVEN HEATING

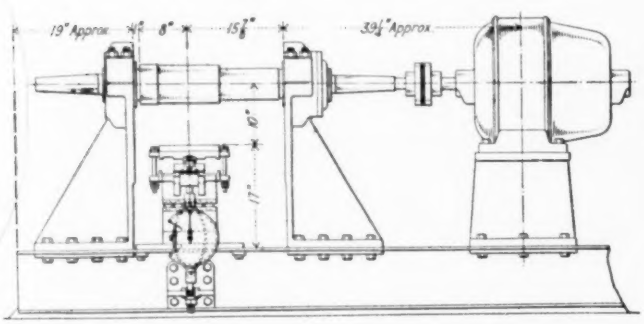

FIG. 16 ARRANGEMENT OF BEARING TESTING MACHINE

of waste-packed bearings, it is evident that a wear of the order of 0.001 in. at the ends must take place before the journal can seat itself uniformly over the whole length of the shell, under running conditions. Before this occurs, load is carried only by the ends of the shell, which accounts for the occasional scoring of the bearings at these places. This fact should be considered in the design of these bearings.

THICKENING OF THE OUTFLOWING OIL

It was noticed that the oil C passing through the bearing and dripping out of its ends was considerably denser than fresh oil put into the oil chamber. The density increased inversely with the rate of flow, the oil attaining a jelly-like state at a rate of flow of one drop in several minutes. Distillation of the lighter constituents from the oil cannot alone account for this. Samples of a commercial car oil both before and after passing through a bearing were examined. Distillation tests under vacuum (1 to 2 mm. pressure) gave the boiling temperatures of both oils after definite fractional distillation[12] as shown in Table 1.

The temperature of 372 deg. fahr. for the used oil would indicate, when compared with data for fresh oil, a loss of approximately 13 per cent due to distillation while passing the bearing. If 20 per cent of this oil is distilled off, the remainder should correspond to $0.80 \times 0.87 = 0.70$ of the new oil, or what remains after 30 per cent of it

[12] This investigation was made by Mr. Wilharm of the Chemical Section of the Westinghouse Research Department.

TABLE 1 BOILING TEMPERATURES OF FRESH AND USED OILS AFTER DEFINITE FRACTIONAL DISTILLATION

Part of oil distilled off, per cent	Boiling temperatures in vacuum, deg. fahr. — Fresh oil	Used oil
0	250	372
5	316	474
10	362	508
15	388	528
20	406	557
25	428	...
30	445	...
35	462	...
45	494	...
55	536	...
65	590	...
75	624	...
85	692	...
95	710	...

is distilled off. Yet the boiling temperatures of the two remainders are 557 deg. fahr. and 445 deg. fahr. There is evidently some oxidation of the oil taking place, and the influence of this on the lubricating action of the oil will be worth while investigating.

Appendix

DESCRIPTION OF THE TESTING MACHINE

THE general arrangement of the bearing testing machine is shown in Fig. 16. A test journal is mounted on self-aligning roller bearings in two pedestals, and a test bearing may be placed on the table underneath the journal and pressed against it with a force of 12,500 lb. In addition, the shaft has on both ends tapered extensions on to which sleeves with tapered bores may be slipped, thus forming two additional journals on which axle

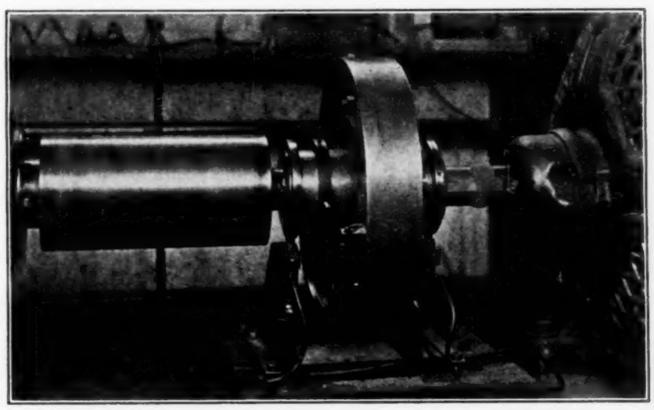

FIG. 17 TORSIOMETER USED IN EXPERIMENTS

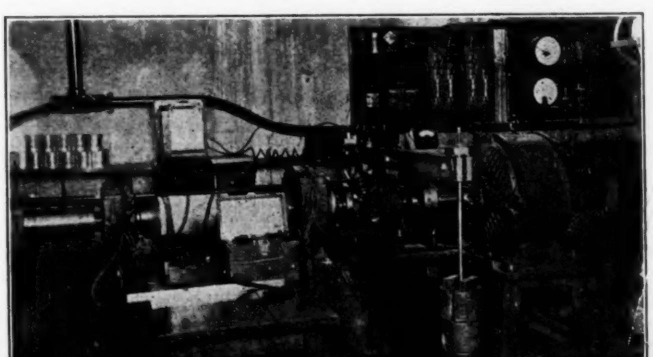

FIG. 18 TESTING EQUIPMENT WITH AXLE BEARING MOUNTED FOR TEST

bearings or similar types requiring low loads may be tested. The journal shaft is driven by a directly connected 12-hp. motor through a flexible coupling which can be replaced by a torsiometer coupling.

The test-bearing table is supplied with two straight ball bearings acting as knife edges, one parallel and the other normal to the center line of the journal. Such a construction enables a uniform pressure of the journal

to be exerted against the shell along the bearing, while by wedging up the table it permits a study of the effects of misalignment. The ball bearings allow the bearing to adjust itself freely to the journal in order that an oil film may build up identical with the film in a fixed bearing and a running loaded journal.

The torsiometer measures the total friction torque on the test shaft, from which the friction of the experimental bearing is obtained by deducting the friction in the roller bearings. The friction in these auxiliary bearings is small when compared with the friction in the test bearing and is known for different loads with a fair degree of accuracy, so that the errors in the observed values of friction are not large.

Fig. 17 shows the torsiometer used in the experiments with axle bearings. The rotation is transferred from the motor to the test shaft through a spiral spring. The driving disk carries an insulated coil and the driven sleeve an adjustable arm with a brush holder, the whole making a rotating potentiometer. A constant voltage is applied to the ends of the coil and a recording voltmeter records the voltage drop between one end of the coil and the sliding brush. The position of this brush is a function of the torque. Careful calibration shows a fair straight-line relation between torque and voltage reading. The centrifugal force on the spring involves a certain correction for speed; this was found by experiment.

A view of the equipment with an axle bearing mounted on one of the sleeves is shown in Fig. 18. In the foreground is a table with a potentiometer and a contactor used with the thermocouples. The recording voltmeter giving a record of the friction torque is seen on the shelf.

Discussion

ALLEN F. BREWER[1] referred to the paragraph in the paper in which the probability of insoluble matter in a so-called straight mineral residual car oil clogging the minute channels in cotton wicking was mentioned. He was interested in the possible action of certain compounded steam-railway car oils under such conditions of test; in other words, what effect, if any, the presence of such a component as lead-fish oil soap would have upon the capillarity of cotton wicking.

Conditions in actual operation, Mr. Brewer said, were frequently akin both in steam and electric railway service, especially with the advent of the gas-electric rail car. It would seem advisable, therefore, to extend the author's most interesting studies to such car oils as were commonly used in journal boxes in the former service. Another point of inquiry was the matter of adequate lubrication. Mr. Karelitz mentioned that bearings had been noticed to run warmer due to lack of oil flow until oil and waste became heated up. Would it not be possible for such conditions to give rise to glazing of the waste in the bearing window? Observation of certain instances of the latter, had led to the impression that lack of lubrication in the face of higher temperatures might have been the responsible factor. Certainly if this were the case, the adoption of "oil-sealed" housings would be a decided advance toward more effective lubrication in such bearings.

The author in conclusion stated that most of the mineral oils on the market had insoluble matter at cold temperatures. When liquefied at elevated temperatures these ingredients were carried through the waste. Experiments had shown that at room temperature cotton or wool waste carried over about one-half of the insoluble matter in one compounded oil; with another compounded oil it had been found that 0.7 of the original amount of insoluble matter was contained in oil passed through a waste-packed armature bearing. As to glazing, it was generally due to dirt penetrating into the bearing. In so far as the waste was not completely dry, it would not glaze at shell temperatures as high as 150 deg. cent.

[1] Mechanical Engineer and Editor of *Lubrication*, The Texas Company, New York. Mem. A.S.M.E.

Power in the Packing Industry

IN A PAPER entitled The Production and Use of Power in the Packing Industry, presented at the Second Mid-West Power Conference, in Chicago, Ill., February 15 to 18, 1927, C. H. Kane said: "The meat-packing industry has long since passed the stage of slaughtering, packing, and rendering of by-products, and we believe it occupies an enviable position in the field of great manufacturing concerns."

Reviewing the growth of the industry, he said that 30 to 40 years ago the boiler equipment consisted of a bank of hand-fired, return-tubular boilers operating at low steam pressure. Belted-type, non-condensing engines were located at various points throughout the plant, with an occasional belted generator for supplying current to a few carbon-filament lamps. Mechanical refrigeration was not known in the industry, the carcasses being chilled instead by means of natural ice.

About 1890 marked the end of these crude methods. Higher steam pressures were introduced, and stoker-fired boilers supplanted hand methods. Gradually the individual electric motor came into use, along with the elimination of the maze of belts, as the compound condensing engine and electric generators began to replace the old clumsy power units. Substantial savings in labor and fuel resulted. In a few cases turbines have been introduced, but the nature of the work and the lack of condensing water in sufficient volume have not made them popular.

Great quantities of both hot and cold water are required, and the heating operations, as well as a large part of the processing, may be done by exhaust steam from the many pumps and prime movers. Large quantities of steam are required for rendering, and recent years have produced the jacketed kettles, melters, and driers, allowing return of condensate to the boilers and consequently saving greatly in cost of operation.

Much thought has been given to the generation of higher steam pressures for the operation of power-driven units, such as refrigerating machines, pumps, etc., with the idea of exhausting at pressures up to 150 lb. per sq. in., thus making it possible to bring the entire requirements of the packing-plant steam processing within the scope of the exhaust-steam system. There might be difficulty in balancing the eight-hour service of the processing equipment against the 24-hour service of the refrigerating machinery, pumps, etc., however.

To illustrate the growth of electric-power usage, the author said that in 1916 the Chicago plant of Swift & Co. tried out the service to the extent of a 2500-kw. demand. Eight years later this had increased to 10,000 kw. demand. The load factor increased from 40 per cent to above 65 per cent. Medium-speed, electrically driven refrigerating machines are taking the place of the older low-speed, steam-driven units for summer operation, the motor installation being such that it is easy to shift to steam for the winter months and use the exhaust for heating purposes.

The following approximate figures on the power requirements for pumping, refrigeration, manufacturing, etc., at the Chicago plant were given:

Steam-driven refrigerating machines	8000 hp.
Combined steam-driven and electric-driven pumping equipment	5000 hp.
Electric power for manufacturing, produced by plant	3000 hp.
Electric power for manufacturing, supplied by central station	12000 hp.

In this particular plant the electric power-generating station is equipped to generate some 6000 kw., and an agreement with the central station provides that the plant shall operate this equipment during peak loads of the central station. In the fall and winter months the plant is operated in parallel with the central-station service. The plant is operated with simple twin Corliss engines exhausting into a common header supplying the exhaust-steam system.

Generally the city water mains are not of sufficient size to accommodate the modern packing plant, therefore pumps are required to boost the pressure. Power is also used to pump from wells, often running to depths of 250 ft. Where air lifts are used, two-stage compressors direct-connected to medium-speed synchronous motors have been found very satisfactory.

In the manufacture of shortening, vegetable oils are solidified by hydrogen treatment, and the large quantities of the gas required are obtained by electrolytic dissociation of water. Direct current at 120 volts is passed through a series of cells requiring 2.2 volts each, the amount of current being from 400 to 600 amperes, depending upon the size of the cell. The electrolyte is caustic potash, and hydrogen is collected at the cathode and oxygen at the anode.

Thawing by electricity is being considered, but at first glance it does not seem to compare favorably with steam; however, when line losses are considered in winter, it becomes more attractive. Further, steam cannot be snapped on and off at will, neither can it be operated automatically with the same degree of simplicity. Experiments are also being carried on to effect a better method of thawing tank cars loaded with oils in winter, and an electric heater seems to meet with favor.

The Reaction of a Nozzle on a Flat Plate

Static and Dynamic Reactions of Jet—Effect of Position of Plate on Discharge of Nozzle—Distance Plate Must Be from Nozzle Tip for Nozzle to Give Full Discharge

By CHARLES BOEHNLEIN,[1] MINNEAPOLIS, MINN.

In March, 1925, Mr. Adolph F. Meyer[2] called the author's attention to the fact that there had been little or no work done on the reactive effect of a jet of water against a flat plate when the flat plate was close to the nozzle tip. It is commonly known that the force of a jet against a flat plate is about equal to the change in momentum of the jet before and after impact when the plate is sufficiently far away from the nozzle tip so that the plate will not influence the discharge of the nozzle. If the plate is against the nozzle so that there is no water discharging, the force necessary to hold the plate in that position is then equal to the product of the static pressure of the water and the area of the nozzle. If the plate is in the first-named position, i.e., relatively far away from the nozzle, and the jet of water deflects through an angle of 90 deg., it can be proved theoretically and verified experimentally that the reaction of the water on the plate is equal to the product of the mass of the water and the velocity of the jet. Neglecting losses in the nozzle, the product becomes double the force required in the second-named position.

Let the force exerted on the plate when the nozzle tip is relatively far away from the flat plate be called the "dynamic reaction," and when the flat plate is against the nozzle tip let that force be called the "static reaction." In these terms the facts stated above are that the dynamic reaction is theoretically double the static reaction. Experimentally it is never quite double since there is always a certain amount of head lost in the nozzle, but in the case of a smooth-bore nozzle whose coefficient of discharge is equal to 0.96, the dynamic reaction on the plate is about 1.84 times the static reaction.

In this paper the effect on the plate has been discussed in two positions, viz., relatively far away from the nozzle tip and against the nozzle tip. The questions now arise: (1) What is the reaction on the plate when it is put in intermediate positions? (2) How great must this relative distance of the plate be in order that the reaction on the plate shall be purely dynamic? (3) Incidentally, what effect has the position of the plate on the discharge of the nozzle, and how far must the plate be from the nozzle tip so that the nozzle is giving full discharge? This was the problem which Mr. Meyer set before the author, and the object of this experiment.

Description of Apparatus

Since the above problem is one that is of importance in hydraulic regulators where the governing element is a nozzle discharging on a flat plate, it was attempted to make the dimensions of the nozzle comparable with those used on such apparatus. Since the water used by the nozzle is waste and it is necessary to screen this water of dirt, it is of importance that the amount be kept small. On the other hand, when the apparatus is small the forces involved are small and the percentage of error is large. So in the first case it was decided to use a $^1/_8$-in. smooth-bore nozzle on the end of a $^1/_4$-in. standard pipe discharging on a $^7/_8$-in.-diameter flat plate. This series of runs was considered more or less as a preliminary set to determine if the apparatus was sensitive enough to obtain the desired degree of accuracy. It was decided that for the second series of runs a $^1/_4$-in. nozzle discharging on a $1^1/_4$-in.-diameter plate would be used. Although the $^1/_8$-in. nozzle gave results sufficiently close for practical purposes, a higher degree of accuracy was desired.

The set-up of the apparatus as used in Test 1 is shown in Fig. 1. The supply water was led into a drum whose diameter is 12 in. and length 20 in. This supply drum was thought necessary so that the velocity head would be so small that the total head would be only the pressure plus the potential head. It was also thought that it would eliminate any air in the water. The water was conducted from the supply drum down the $^1/_4$-in. pipe and discharged through a smooth-bore nozzle against a flat plate. The $^1/_4$-in. pipe and the nozzle were in a vertical position, as shown in Fig. 1. The mercury manometer was connected to the supply drum and to three points along the $^1/_4$-in. pipe as shown in Fig. 1. At A, B, C, and D are stopcocks so that the respective pressures at these different points could be measured independently by one manometer. The discharge from the nozzle was collected by receiving pans which surround the plate and nozzle, and the water was then conveyed to the

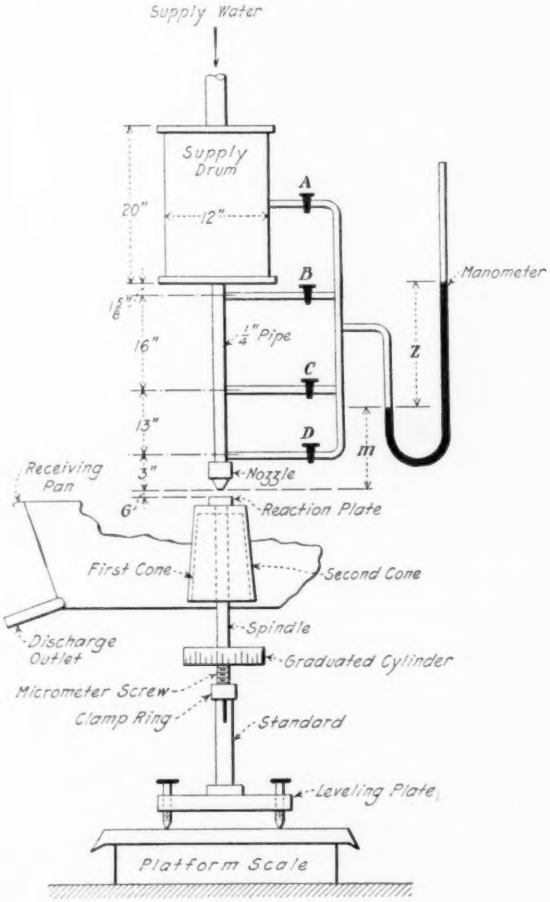

Fig. 1 Set-Up of Apparatus as Used in Test 1

weighing tank. The receiving pans were not connected mechanically to the spindle which supported the reaction plate, but each pan contained an open cone, like a cake pan. Attached to the spindle was a second cone which telescoped over the first cone. This second cone acted as an umbrella to the cone on the pan (see Fig. 1). During the experiment the utmost precaution was taken to see that the two cones did not touch. This arrangement was easy to construct and no water leaked through on the scales. All measurements of discharge were made by weighing the water over a given length of time.

The spindle to which the reaction plate was attached had, on its lower end, a micrometer screw for the purpose of adjusting the gap G, or the distance in inches or decimals of an inch between the nozzle tip and reaction plate. The reading of this gap was taken on the graduated cylinder. The vertical standard carried the internal thread for the micrometer screw and rested on the table of a platform scale. The scale measured the reaction on the plate.

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Fig. 2 shows a photograph of the apparatus as it was set up in the second series of runs, i.e., Tests 2, 3, and 4. Referring to the numbers on the photograph, (1) shows the standard resting on the platform scale; (2) the receiving pans which collect the water as it discharges off the flat plate; (3) the pipe which conveys the water to the nozzle; (4) the swinging pipe for diverting the water into or out

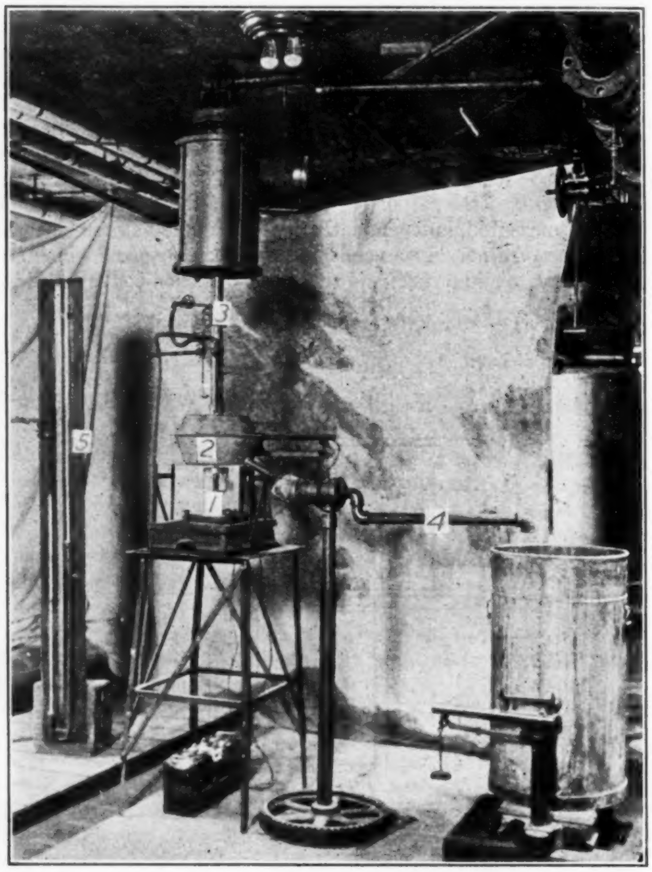

FIG. 2 GENERAL VIEW OF APPARATUS AS ASSEMBLED FOR TESTS 2, 3, AND 4
(1, Standard. 2, Receiving pans. 3, Flange couplings. 4, Pipe for diverting discharge. 5, Mercury manometer for measuring pressures.)

of the weighing tank; and (5) the manometer for measuring the pressures.

The pressure was measured by an ordinary open U-tube mercury manometer as shown in Fig. 2. The manometer had a brass scale graduated to hundredths of a foot, so that measurements of pressure could be estimated to the one-thousandth part of a foot of mercury. The connections to the $^1/_4$-in. pipe, or piezometer rings, were made by screwing close nipples into the run of a $^3/_8$-in. tee, and drilling out the nipples until they would slip over the $^1/_4$-in. pipe. The nipples were then soldered in position as shown in Fig. 3. Previous to placing the tee in position, four $^1/_{16}$-in. holes were drilled into the $^1/_4$-in. pipe. Fig. 4 shows the small brass nozzle used in Test 1. This nozzle has been referred to as a $^1/_8$-in. nozzle, but its actual bore was 0.128 in. It was attempted to make the nozzle tip as sharp-edged as possible; but it was of course necessary to have a flat surface around the tip to protect the bore from mechanical damage. The nozzle tip was not touched except to obtain the zero reading of the gap.

The standard and micrometer screw to which was attached the reaction plate, shown in Fig. 1 and in (1) on Fig. 2, is also shown in a close-up view in Fig. 5. The micrometer screw was machined in a lathe to $^1/_2$ in. in diameter, 20 threads per inch. The standard was threaded internally with a tap. The upper part of the standard was split across two diameters like a chuck and a tapered thread placed on the outside. A clamp ring with an internal tapered thread was screwed over the external tapered thread on the standard and tightened, thus squeezing the internal micrometer thread tightly against the screw on the spindle. The purpose of this was to take up the backlash in the screw. The pitch of the micrometer screw was checked against the vertical screw of a milling machine and no variation was found. As a result of this check the pitch was found to be accurate to 0.0001 in. for $^1/_2$ in. of the screw length. The screw carried a graduated dial on Test 1 and cylinder on Tests 2, 3, and 4. In Test 1 a dial 2 in. in diameter with 50 divisions was used. This made each division equal to 0.001 in. on the gap. The first dial was placed in the receiving pans and was difficult to read and almost impossible to estimate to 0.0001 in. while in operation. There are no illustrations showing this first arrangement. On Tests 2, 3, and 4 a cylinder 4 in. in diameter and placed below the receiving pans, as shown in Fig. 2, was divided also into 50 divisions, which made it easy to estimate the gap reading to 0.0001 in.

In Fig. 5 the standard can be seen resting on the platform scale. It will be observed that there is a square cast-iron plate upon which the standard rests. This plate has three set screws, one at each corner on the front edge, and the third behind the standard. These screws were used for leveling the reaction plate on the top of the spindle.

The scale upon which this apparatus rested was a small platform balance capable of weighing a load of 300 lb. The beam of this scale was graduated to hundredths of a pound. The ratio of the scale leverage was 40 to 1, so that if the end of the beam was swung through a distance of 1 in., the platform moved through a distance of $^1/_{40}$ in. This, of course, would change the gap too much since it was desired to measure that distance closer than 0.001 in. To overcome this the end of the scale beam was fitted with two electrical contacts, one above and one below, as shown in Fig. 6. The travel of the beam was restricted to 0.004 in. of movement. This restricted the platform movement to 0.0001 in. When the beam was up against the upper contact the upper lamp would light; when down, the lower lamp would burn. Even with this restricted movement of the beam, the scale was apparently very sensitive to $^1/_{200}$ lb. when there was about 18 lb. on the platform and showed indication when a 1-gram weight was placed on the platform. However, the author does not believe that the readings were any closer than ± 0.005 lb.; and in runs 1 to 51

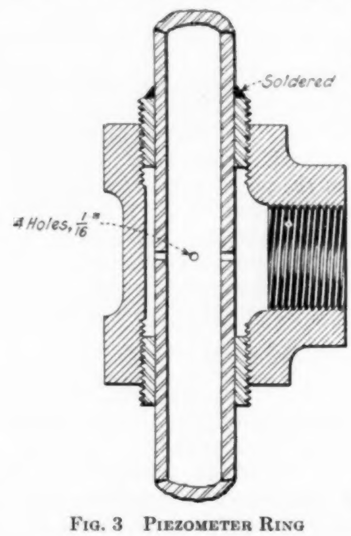

FIG. 3 PIEZOMETER RING

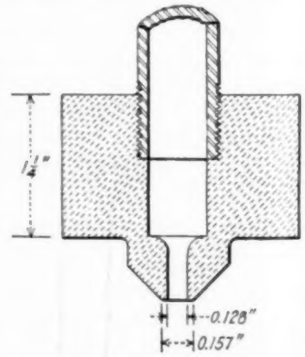

FIG. 4 SMALL BRASS NOZZLE USED IN TEST 1

it is apparent that some readings contained an error as high as 0.02 lb. The scales were calibrated twice during the runs against standards here in the laboratory.

With the apparatus above described, the author believes that the percentage of error in the discharge was within 1 per cent. On small rates of discharge the runs were made 20 min. and over. On larger rates of discharge the time was never below 3 min. Most of the readings, with the exception of the reaction in runs 1 to 51, inclusive, show an accuracy within 1 per cent, as one will observe by inspection of the curves.

In the second series of tests a nozzle with a bore of 0.251 in., referred to above as a $^1/_4$-in. nozzle, was used. This nozzle can be seen in Figs. 2, 5, and 7. Fig. 7 shows the details of the set-up as well as some of the dimensions. The pipe to which the nozzle is attached is a standard $1^1/_2$-in. wrought pipe. Between the drum and the nozzle will be observed in Fig. 2 and also in Fig. 7 a flange coupling. At this place an orifice plate was located to cause a certain amount of resistance to the flow of water. Above the upper flange can be seen in Fig. 7 a piezometer ring which connects through

a globe valve to the manometer. At the base of the nozzle will be noticed in the figure a second piezometer ring which also communicates with the manometer through a globe valve. The purpose of the globe valves was to allow the pressure at either ring to be measured independently by one manometer. The pressure above the orifice the author has termed the "supply pressure;" that at the base of the nozzle, the "chamber pressure." It is the chamber pressure that does the work of operating the main valve or the relay valve in hydraulic regulators.

Development of Formulas

The exact theory of this experiment is a problem in hydromechanics and is of course very complicated, but can be simplified if the usual approximations which are used in hydraulics are employed. Throughout the development of the formulas the following notation is used:

Section 1 designates the section of the piezometer ring above the orifice (see Fig. 7); this corresponds to A in Fig. 1

Section 2 designates the section at the piezometer ring at the base of the nozzle (see Fig. 7); this corresponds to D in Fig. 1

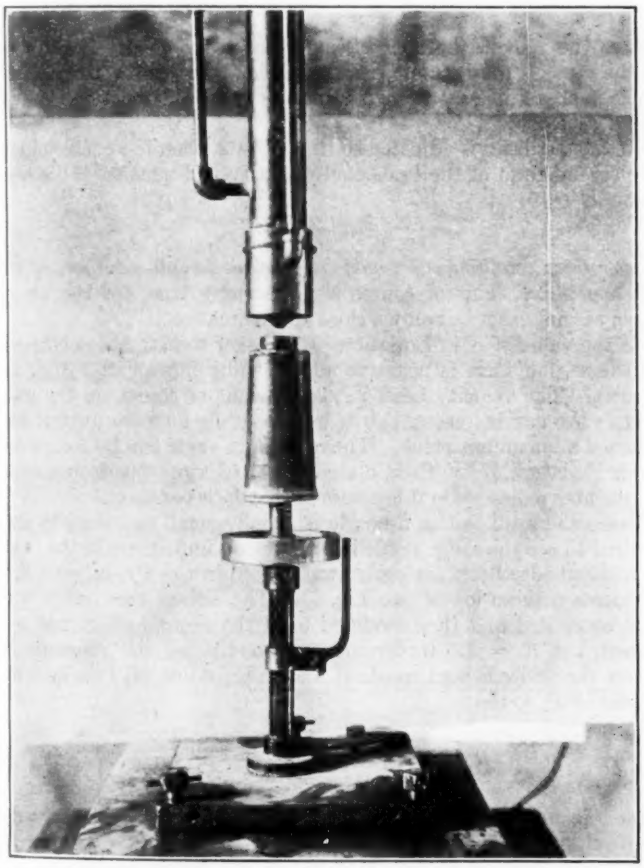

FIG. 5 CLOSE-UP VIEW WITH RECEIVING PANS REMOVED, SHOWING NOZZLE, REACTION PLATE, SPINDLE, STANDARD, AND LEVELING PLATE RESTING ON PLATFORM SCALE

Section 3 designates the section at the nozzle tip, or just inside in the straight part of the bore.

A_1 = area at section 1, sq. ft.
D_1 = diameter at section 1, ft.
P_1 = pressure at section 1, lb. per sq. ft., termed the supply pressure for Tests 3 and 4
V_1 = velocity at section 1, ft. per sec.
a = area of orifice, sq. ft.
d = diameter of orifice, ft.
C_v = coefficient of velocity for orifice
C_c = coefficient of contraction for orifice
C = coefficient of discharge for orifice
V = actual velocity of efflux from orifice, ft. per sec.
A_2 = area at section 2, sq. ft.
D_2 = diameter at section 2, ft.
P_2 = pressure at section 2, lb. per sq. ft., termed the chamber pressure for Tests 3 and 4[3]
V_2 = velocity at section 2, ft. per sec.

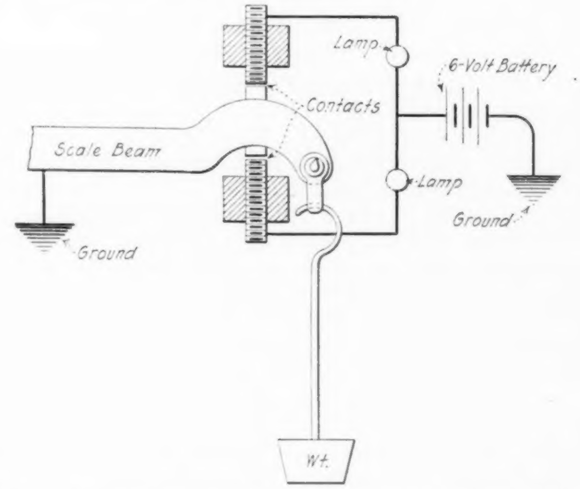

FIG. 6 ARRANGEMENT OF ELECTRICAL CONTACTS ON SCALE BEAM

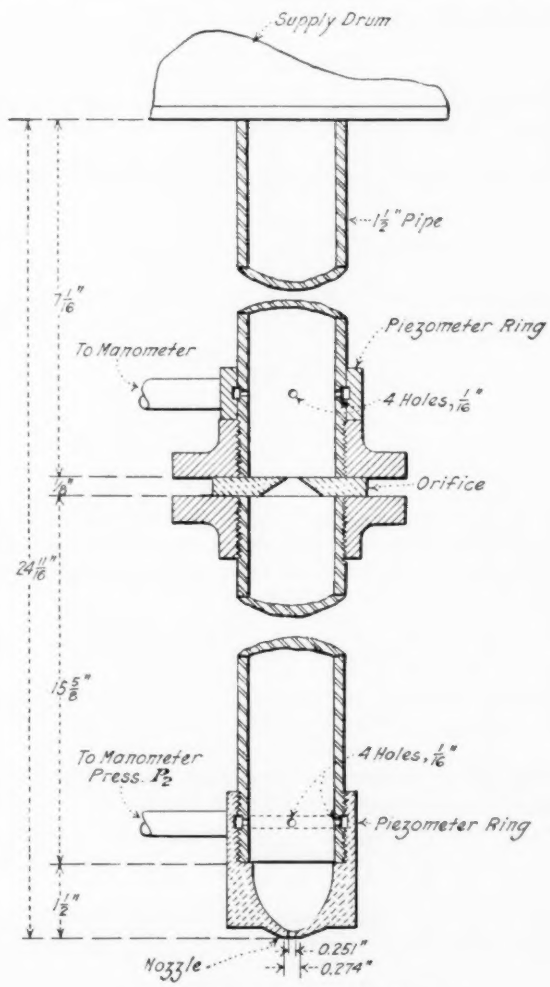

FIG. 7 DETAILS OF NOZZLE SET-UP

A_3 = area of nozzle bore, section 3, sq. ft.
D_3 = diameter of nozzle bore, section 3, ft.
P_3 = pressure at section 3, lb. per sq. ft.
V_3 = actual velocity at section 3, ft. per sec.
C_{v3} = coefficient of velocity for nozzle
C_{c3} = coefficient of contraction for nozzle

[3] P_2 was the supply pressure for Test 2 and there was not any intermediate or chamber pressure in this test.

C_3 = coefficient of discharge for nozzle
Q = discharge, cu. ft. per sec.
M = mass of water per sec.
R = reaction on plate, lb.
w = density of water = 62.5 lb. per cu. ft.
g = acceleration due to gravity = 32.2 ft. per sec. per sec.

P_1 and P_2 in the above notation are the pressures corrected to the datum plane tangent to the surface of the reaction plate. If h is the vertical distance between the datum plane and the piezometer ring and if P' is the pressure at the higher elevation, then P_1 or P_2, as the case may be, is equal to $P' + wh$, where P' and h are the respective values for P_1 or P_2. All pressures in this experiment are corrected to the above-named datum plane or to a plane through the nozzle tip, which amount to the same thing since the added potential head due to the lower elevation of the reaction plate is practically negligible compared with the total head used in this experiment.

When the reaction plate is against the nozzle tip it is evident that

$$R = A_3P_3 \qquad [1]$$

When the plate is sufficiently far away to develop full dynamic reaction, the equation becomes

$$R = MV_3 \qquad [2]$$

In all intermediate positions it is assumed that

$$R = A_3P_3 + MV_3 \qquad [3]$$

Attention is called to the fact that P_3 is not the full static pressure of the supply drum, nor is it atmospheric, but is some value between these two.

The relation between mass and discharge of water per second is

$$M = \frac{Qw}{g} = \frac{wA_3V_3}{g} \qquad [4]$$

The second part of this expression is true only if the coefficient of contraction is equal to unity. If [4] is substituted in [3] it becomes

$$R = A_3P_3 + wA_3\frac{V_3^2}{g}$$

or

$$R = wA_3\left(\frac{P_3}{w} + \frac{V_3^2}{g}\right) \qquad [5]$$

P_3/w is of course the pressure head in feet of water and may be computed if the pressure at the base of the nozzle is known. If Bernoulli's theorem is written between the base of the nozzle and the nozzle tip, sections 2 and 3, then,

$$\frac{V_2^2}{2g} + \frac{P_2}{w} = \frac{V_3^2}{2g} + \frac{P_3}{w} + \frac{V_3^2}{2g}\left(\frac{1}{C_{v3}^2} - 1\right)$$
$$= \frac{V_3^2}{2g}\frac{1}{c_{v3}^2} + \frac{P_3}{w} \qquad [6]$$

By making use of the theorem of continuity, i.e., $V_2A_2 = V_3A_3$, and solving for P_3/w from Equation [6], there results

$$\frac{P_3}{w} = \frac{P_2}{w} - \frac{V_3^2}{2g}\left(\frac{1}{C_{v3}^2} - \frac{A_3^2}{A_2^2}\right) \qquad [7]$$

If the quantity within the parenthesis is represented by K—and it is apparent that K is a constant—Equation [7] may be written

$$\frac{P_3}{w} = \frac{P_2}{w} - \frac{V_3^2}{2g}K \qquad [8]$$

If the gap is made sufficiently large so that P_3 is atmospheric pressure, then K can be computed from Equation [8], since P_2 was measured by the manometer and V_3 is the quotient of Q divided by A_3 provided C_{v3} is equal to 1. It is to be noted that K is independent of the gap since it only represents the loss to the nozzle tip and is not dependent on the losses between the edge of the nozzle and the reaction plate. If the value of P_3/w from Equation [8] be substituted in Equation [5], then

$$R = wA_3\left[\frac{P_2}{w} + \frac{V_3^2}{2g}(2 - K)\right] \qquad [9]$$

Equation [9] is general, for it makes no difference where P_2 is measured since P_3 can be computed from Equation [8], except that the constant K will be different for the different positions along the pipe.

Since

$$K = \frac{1}{C_{v3}^2} - \frac{A_3^2}{A_2^2}$$

and if the area A_2 is large compared with A_3 then $K = 1/C_{v3}^2$, and if $C_{v3} = 1$, which is the case if there is no loss, then $K = 1$; and under these conditions

$$R = wA_3\left[\frac{P_2}{w} + \frac{V_3^2}{2g}\right]$$

If the reaction plate is sufficiently far away so that P_3 = atmospheric pressure, then the velocity head $V_3^2/2g$ would be equal to P_2/w, hence

$$R = 2A_3P_2$$

where the velocity head has been replaced by P_2/w. The above is the case of full dynamic reaction if there were no lost head; but since in an actual case there is some lost head, the reaction is never quite $2A_3P_2$.

If the lost head is adjusted so that $K = 2$, then the coefficient of the second term of the bracketed quantity in Equation [9] is zero and

$$R = A_3P_2$$

Under these conditions R would be constant for all positions of the reaction plate. This of course is not strictly true, for the above theory is not exact but only a close approximation.

If the value of K in Equation [9] is larger than 2, the coefficient of the second term is negative and R will diminish as $V_3^2/2g$ increases. The velocity head $V_3^2/2g$ depends of course on the gap, and as the gap increases, $V_3^2/2g$ will naturally increase until it has reached a maximum value. The maximum value can be computed from Equation [7] if P_3 is placed equal to atmospheric pressure. In the above discussion it is assumed that P_2 is constant.

Since the head lost in a nozzle is usually small and since it was desired to use the same nozzle in all tests, an impedance in the form of a sharp-edged circular orifice was placed in the $1^1/_2$-in. pipe line to increase these losses (see Fig. 7). The size of this orifice was first computed and then modified until the desired effect was obtained, i.e., $K = 2$. Under the new conditions, the pressure P_1 above the orifice is kept constant, and in Equation [9] P_2 is now replaced by P_1 so that

$$R = wA_3\left[\frac{P_1}{w} + \frac{V_3^2}{2g}(2 - K)\right] \qquad [10]$$

K now includes all losses from the nozzle tip to the piezometer ring above the orifice.

In order to compute the diameter of the orifice the losses were summed up as follows:

Total loss of head = loss in orifice + loss due to sudden enlargement + loss in nozzle

The formula used to compute the loss of head in the orifice is

$$\text{Lost head in orifice} = \frac{V^2}{2g}\left(\frac{1}{C_v^2} - 1\right)$$

The same formula can be used for the nozzle if the values of V and C_v pertain to the nozzle. The formula used to compute the loss due to sudden enlargement is

$$\text{Lost head} = \frac{(V - V_2)^2}{2g}$$

The above loss occurs where the orifice discharges into the $1^1/_2$-in. pipe. Combining these losses and making use of the theorem of continuity, the formula becomes

$$\text{Total lost head} = \frac{V_3^2}{2g}\left\{\frac{D_3^4}{D_1^4}C_{c3}^2\left[\frac{D_1^4}{d^4}\left(\frac{1}{C^2}-\frac{1}{C_c^2}\right)+\left(\frac{D_1^2}{d^2C_c}-1\right)^2\right]+\frac{1}{C_{v3}^2}-1\right\}$$

Attention is called to the fact that $D_1 = D_2$ (see Fig. 7) and also that the nozzle is assumed to have a coefficient of contraction. If the above quantity within the braces is represented by the Greek letter β, then,

$$\text{Total lost head} = \frac{V_3^2}{2g}\beta \qquad [11]$$

If Bernoulli's theorem is written between sections 1 and 3 and solved for P_3/w, this becomes

$$\frac{P_3}{w} = \frac{P_1}{w} + \frac{V_3^2}{2g}\left(\frac{C_{c3}^2D_3^4}{D_1^4} - 1 - \beta\right) \qquad [12]$$

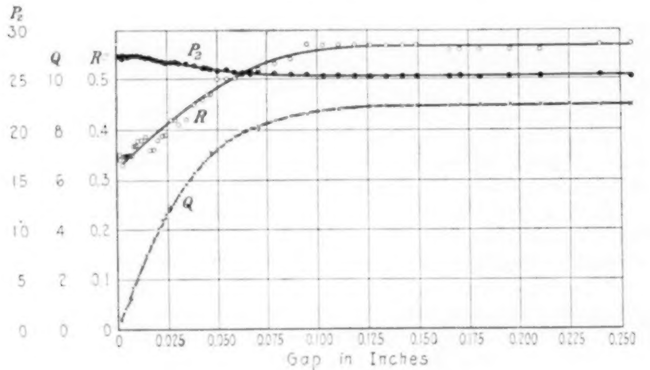

FIG. 8 CURVES GIVING RESULTS OF TEST 1
(P_2 = pressure at base of nozzle, lb. per sq. in. Q = discharge, cu. in. per sec. R = reaction on plate, lb.)

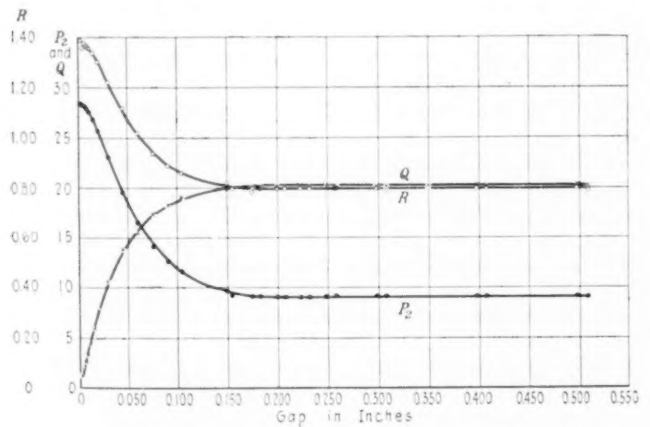

FIG. 10 CURVES GIVING RESULTS OF TEST 3
(R = reaction on plate, lb. P_2 = chamber pressure, lb. per sq. in. Q = discharge, cu. in. per sec.)

Substituting the above value of P_3/w in Equation [5], there results

$$R = wA_3\left[\frac{P_1}{w} + \frac{V_3^2}{2g}\left(\frac{C_{c3}^2D_3^4}{D_1^4} + 1 - \beta\right)\right] \qquad [13]$$

If Equation [13] is compared with Equation [10], it will be observed that

$$2 - K = \frac{C_{c3}^2D_3^4}{D_1^4} + 1 - \beta$$

Solving this equation for K and substituting for β its original expression, then

$$K = \left\{\frac{D_3^4}{D_1^4}C_{c3}^2\left[\frac{D_1^4}{d_4}\left(\frac{1}{C^2}-\frac{1}{C_c^2}\right)+\left(\frac{D_1^2}{d^2C_e}-1\right)^2\right] + \frac{1}{C_{v3}^2} - \frac{D_3^4}{D_1^4}C_{c3}^2\right\} \qquad [14]$$

Since the velocity head in the 1½-in. pipe was small, the loss due to pipe friction was neglected and does not enter into the above equation.

In order to compute the diameter of the orifice, it is necessary to know the coefficients for both the nozzle and the orifice. The coefficients for the nozzle were computed from the data contained in Test 2. From the data from which Fig. 9 was plotted it is found that the reaction and the discharge have reached their maximum values when the gap is 0.2253 in. The coefficients for the orifice were taken from A. H. Gibson's Hydraulics, third edition, page 111. With the following data the diameter of the orifice was computed.

$K = 2$	$C = 0.628$
$D_1 = 1.577$ in. measured value	$C_{v3} = 0.974$
$D_3 = 0.251$ in. measured value	$C_{c3} = 0.920$
$C_c = 0.641$	

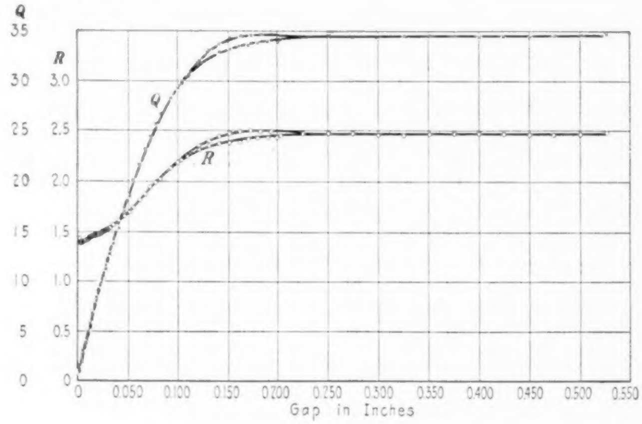

FIG. 9 CURVES GIVING RESULTS OF TEST 2
(Q = discharge, cu. in. per sec. R = reaction on plate, lb.)

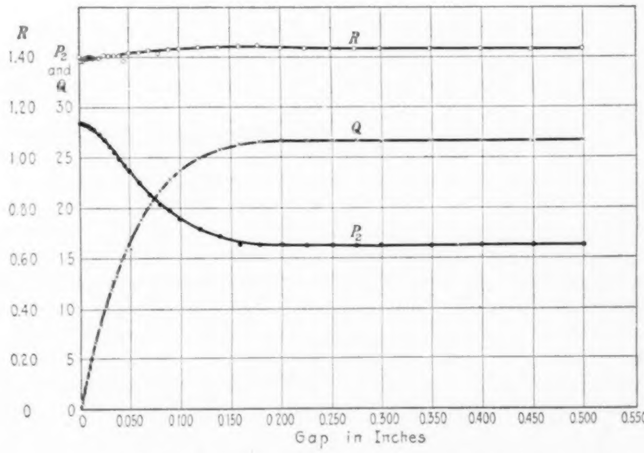

FIG. 11 CURVES GIVING RESULTS OF TEST 4
(P_2 = chamber pressure, lb. per sq. in. Q = discharge, cu. ft. per sec. R = reaction on plate, lb.)

The diameter was found to be 0.26 in. This diameter proved to be too small.

DISCUSSION OF RESULTS

As mentioned before, Test 1 was a preliminary experiment more for the purpose of studying the apparatus than for the results. In this test it was attempted to keep the pressure in the supply drum constant. Together with the reaction and discharge the pressure was measured at sections A, B, C, and D (see Fig. 1). The results of this test are shown in the curves of Fig. 8.

Test 2 was made with a larger nozzle as shown in Fig. 2. The orifice shown in this figure was omitted; as a matter of fact, the pipe was cut in two and fitted with the flange coupling after Test 2 had been completed. In this test it was attempted to keep the pressure at the base of the nozzle constant. The results of the above test are shown in the curves of Fig. 9.

Test 3 was made with an orifice plate having an opening of 0.280

in. between the flanges of the coupling, as shown in Fig. 7. In this test the pressure at the piezometer ring above the orifice was kept constant. Results are shown in the curves of Fig. 10.

Test 4 was the same as Test 3 except that the orifice diameter was 0.330 in. Fig. 11 pertains to Test 4.

Below is given an outline of the experiment as it was performed.

OUTLINE OF EXPERIMENT

Test No.	Arrangement shown in	Curves shown in	Constant pressure at section	Value of constant pressure[1]	Impedance
1	Fig. 1	Fig. 8	*A*	27.37	1/4 in. pipe
2	Fig. 7	Fig. 9	2	28.54	None
3	Fig. 7	Fig. 10	1	28.50	0.280-in-diam. orifice
4	Fig. 7	Fig. 11	1	28.50	0.330-in-diam. orifice

[1] Defined as supply pressure.

The water supply pressure used in this experiment was produced by a centrifugal pump direct-connected to an alternating-current induction motor, and as a result of slight variations in motor speed the pressure did not stay constant but was subjected to small variations from run to run, and even during a single run. In order to bring about more uniformity of results each test was corrected to some standard constant pressure as shown in the above outline. This value of constant pressure given above is the average for that test, except in Test 3, which was corrected to the average of Test 4 since the averages of these two tests were nearly equal. The formulas for making the corrections were developed as follows:

It is apparent that the discharge velocity between the reaction plate and the nozzle tip will be proportional to the square root of the supply pressure. It is also evident that with any given setting of the gap there will be a constant relation between the above-named discharge velocity and the velocity in the nozzle bore (V_3), so that

$$V_3 = b\sqrt{2g\frac{P}{w}}$$

or

$$\frac{V_3^2}{2g} = b^2\frac{P}{w} \qquad [15]$$

where b is an overall coefficient and takes care of the constant relation between the discharge velocity and the velocity in the nozzle bore. P is the supply pressure.

From Equation [15] an equation for Q can be written, as follows:

$$Q = bA_3\sqrt{2g\frac{P}{w}}$$

If letters with the subscript o are used to represent observed values and the notation without the subscript to represent the corrected values, then from the above equation it is evident that

$$\frac{Q}{Q_o} = \frac{\sqrt{P}}{\sqrt{P_o}}$$

In order to correct for the reaction (R) let the velocity head given in Equation [15] be substituted in Equation [9] or [10], and if P_1 or P_2, which designate respectively the supply pressures in [9] or [10] be written without subscripts, then

$$R = A_3w\left[\frac{P}{w} + b^2\frac{P}{w}(2 - K)\right]$$

$$= A_3P\,[1 + b^2(2 - K)]$$

The bracketed quantity in the above equation is, for any setting of the gap, constant, and consequently the following proportion will be true:

$$\frac{R}{R_o} = \frac{P}{P_o}$$

By means of Equation [15] it can be proved in like manner that the chamber pressures of Tests 3 and 4 have the following proportion:

$$\frac{P_2}{P_{2o}} = \frac{P}{P_o}$$

where P_{2o} = observed value of P_2 and
P_2 = corrected value.

To clarify the above formula, run 132, Test 3, is taken as an example. In run 132—

Q_o = 7.68 cu. in. per sec.
R_o = 1.31 lb.
P_{2o} = 25.92 lb. per sq. in.
P_o = 28.63 lb. per sq. in. (observed supply pressure).

In Test 3 the results were corrected to a supply pressure of 28.50 (see outline). Then,

$$\frac{Q}{7.68} = \sqrt{\frac{28.50}{28.63}} \text{ or } Q = 7.67 \text{ cu. in. per sec.}$$

$$\frac{R}{1.31} = \frac{28.50}{28.63} \quad \text{or } R = 1.30 \text{ lb.}$$

$$\frac{P}{25.92} = \frac{28.50}{28.63} \quad \text{or } P = 25.80 \text{ lb. per sq. in.}$$

The above corrections in all cases were small, the largest difference between the observed and corrected values being less than 2 per cent.

Referring again to Test 1, as was pointed out before, the pressure in the supply drum was held constant as the gap was increased. In runs 1 and 2 the reactions are 0.34 and 0.35 lb., respectively, when the gap is 0.001 in. The reaction gradually increases up to 0.57 lb. with a gap of 0.094 in., and for the remaining runs fluctuates between 0.57 and 0.56 lb. It is to be noticed that the reaction has an increasing or rising characteristic as the gap increases. In Fig. 8, which perhaps rounds off experimental error to some extent, it will be observed that the reaction curve was drawn as a straight horizontal line from a gap about equal to the bore of the nozzle (0.128 in.). Owing to the short duration of each run (about 5 minutes) and the fact that there were many readings to be taken, the pressures were read only once each run. The pressure was, without doubt, the largest source of error in the whole experiment, for in some runs the pressure varied as much as 0.100 ft. of mercury. The computed reaction was obtained by the use of Equations [1] and [2]. In Equation [1] P_3 was computed from Equation [8], and K in Equation [8] was obtained from the data given in runs 41 to 51, inclusive. In the above computation the coefficient of contraction was not considered, and this may account for the fact that the computed reaction is larger than the observed. The discharge is of course zero when the reaction plate is against the nozzle tip, but increases very rapidly as the gap is increased. The discharge, however, approaches the constant value rather slowly and it becomes fairly constant when the gap is 0.126 in. (run 40). The pressure at the base of the nozzle, which gradually falls off as the gap increases, also becomes more or less constant at run 40. The gap at run 40 is nearly equal to the nozzle bore.

Tests 2, 3, and 4 were with nozzle of bore 0.251 in. A different shape was used in an attempt to increase the coefficient of discharge. This, however, was not the case since the small nozzle had a discharge coefficient of 0.94 against 0.897 for the larger nozzle. In Test 2 the pressure at the base of the nozzle was kept constant as the gap was changed. Since the equipment was modified for this test, the work was carried out to a higher degree of accuracy than in Test 1. The reaction of run 52 is 1.434 lb. with a gap of 0.0010 in. If the reaction is computed by Equation [1] it will be 1.412 lb., which is slightly less than the observed reaction. The larger observed value was perhaps due to the fact that the pressure acted upon a little larger area than the bore of the nozzle. It is also to be observed that after the first reading in Test 2, as well as in Test 4, the reaction drops slightly and then begins to increase again. This was attributed to the fact that the nozzle tip was not sharp-edged but had a certain amount of width, so that the phenomenon observed is that of water flowing radially outward between two parallel plates. This phenomenon causes a certain amount of suction, depending upon the velocity of the water and the diameter of the plates. As the gap becomes larger the water breaks away from the flat surface on the nozzle tip, and this phenomenon is no longer present. It is to be noticed in Test 3 where the reaction has a falling characteristic that in the second reading (run 127) the reaction drops to 1.36 lb. and then returns to 1.38 lb. before it drops off

farther (see Fig. 10). This phenomenon was also observed on several trial runs which were not recorded.

The discharge of Test 2 is practically the same as that of Test 1, but it is to be observed that the curve (see Fig. 9) shows two values of the discharge from a gap of 0.100 to 0.225 in. The reaction has also two values in this region. The explanation of this may be in the critical velocity effect in the $1^1/_2$-in. pipe. The upper part of the curve was obtained with the gap increasing and the lower with the gap decreasing. The critical velocity was computed by the Osborne Reynolds formula which gave a high value of 0.84 ft. per sec. and a low value of 0.19 ft. per sec. The velocity of the water in the pipe for run 86 was equal to 1.27 ft. per sec., and for run 96 to 1.47 ft. per sec. Both of these velocities are higher than the critical velocities, but the fact that the path of the water at the nozzle was convergent may have increased the values of the critical velocities. On Tests 3 and 4 the critical-velocity phenomenon was not noticed. The disturbance caused by the orifice in the pipe, perhaps, would not permit streamline flow even at low velocities.

In Test 3 an orifice was placed in the $1^1/_2$-in. pipe line. The computed orifice (0.26 in. diameter) was found to be too small, so the runs of that test were discarded. Although the 0.280-in-diameter orifice was also too small it was decided to show these runs, thinking it might be of interest to observe the effect when too much impedance was placed in the pipe, i.e., $K>2$. The results of this test are shown in Fig. 10. Here the reaction decreases as the gap increases or the reaction has a falling characteristic. The pressure at the base of the nozzle (chamber pressure) also falls off rapidly as the gap increases. It is to be remembered that the supply pressure above the orifice is kept constant or nearly so for all setting of the gap.

The next orifice placed in the $1^1/_2$-in. pipe had a diameter of 0.335 in. The results with this orifice are not shown here. The initial reaction on this test was 1.42 lb. with a gap of 0.0010 in.—when the gap was 0.2900 in. the reaction was 1.48 lb. The reaction with this combination gave a slightly rising characteristic.

In Test 4 an orifice 0.330 in. in diameter was used and the results of this test are given in Fig. 11. The lowest value of the reaction was 1.385 lb. and the highest 1.445 lb., which gives a difference of 0.06 lb. The reaction in this test was very nearly constant. The chamber pressure, i.e., pressure at the base of the nozzle, drops from 28.50 lb. per sq. in. to about 16.33 lb. per sq. in., with a difference here of 12.17 lb. per sq. in.

Below are tabulated some of the characteristic values of this experiment.

Test No.	Initial reaction, lb.	Smallest reaction, lb.	Largest reaction, lb.	Difference, lb.	Per cent based on initial reaction	Initial chamber pressure (Lb. per sq. in.)	Final chamber pressure (Lb. per sq. in.)	Difference chamber pressure (Lb. per sq. in.)	Per cent based on initial pressure
1	0.34	0.33	0.57	0.24	71	27.28	25.28	2.00	7
2	1.434	1.383	2.501	1.118	78	...	...	...	..
3	1.39	0.79	1.39	0.60	43	28.43	9.11	19.32	68
4	1.435	1.385	1.445	0.06	4	28.50	16.35	12.15	43

The percentages of change in the reactions and the pressures are based upon the initial readings. In Test 3 it is to be remembered that the smallest reaction (0.79 lb.) is the final reaction and the largest reaction is the initial reaction; this would make the percentage change in reaction negative. For the chamber pressure the initial and final runs were taken. This is not perhaps correct for the final value, but gives a good idea of the amount of change.

Conclusion

From the foregoing experiments the following conclusions were drawn:

1 That with a constant supply pressure the reaction changes as the gap increases. If the resistance between the point of constant supply pressure and the nozzle tip is small, then the reaction increases as the gap increases from its zero position. If the resistance is large the reaction diminishes and the resistance can be adjusted so that the reaction is practically constant for all values of the gap.

2 That the discharge increases up to a maximum value and then remains constant as the gap is increased from the zero position.

3 That the chamber pressure gradually diminishes to a constant minimum value as the gap is increased from its zero position.

4 That the reaction, the discharge, and the chamber pressure reach constant values when the gap is about equal to the diameter of the nozzle bore, and remain constant as the gap is further increased.

Molding Rubber with Electricity—A New Electrolytic Process

MAN has long used rubber to keep electricity where it belongs. Recently some research chemists have been showing how to use electricity to put rubber where man would like to have it. Incidentally, much power-consuming, cumbersome equipment for forming certain kinds of rubber goods to prepare them for vulcanization appears to be doomed to scrap heaps, and some disagreeable work places will be transformed. Patient scientific research, utilizing modern knowledge of the way matter is put together, has again been richly rewarded. Again research is revolutionizing an art; rather, a new art is being built which will in part supplant old methods and in part create new satisfactions for human needs.

Rubber is not soluble in water, but latex and some other forms of pure and compounded rubber can be suspended in water by dispersing them into very minute globules. These "suspensions" or "dispersions" for some purposes resemble a solution, although in other respects are quite different. In water dispersions the globules of rubber carry negative charges of electricity; consequently when a current is passed through them the minute bits of rubber go to the positive pole, or anode.

In 1922, S. E. Sheppard and L. W. Eberlin, working in a laboratory of the Eastman Kodak Company, at Rochester, New York, and Paul Klein, A. Szegvari, and a large staff in Budapest, Hungary, found independently that pure and compounded rubber could be electrolytically deposited from water dispersions on to metal or ceramic molds, in a way somewhat like electroplating with metals. The former group was trying to cover metals with adhering coats of rubber deposited from water dispersions and the latter to make rubber goods directly from latex.

These men conceived the plan of so depositing rubber on forms, from latex, that the rubber particle itself was not altered, and made the astonishing discovery that electrodeposited rubber had the highest quality ever observed. Many were the problems to be solved before this was made practical. It was necessary to incorporate other substances to be deposited simultaneously on the anode, such as sulphur, zinc oxide, and carbon black. These latter two substances are necessary in rubber goods to give toughness. A long investigation was carried out to find means to disperse them in water, mixed with the latex particles, without coagulating the latex. A noteworthy achievement, which has made the anode process possible, has been the means by which these rubber layers may be free from bubbles, for when an electric current is passed through a solution, bubbles of gas are formed on anode and cathode. So they hit upon the scheme of surrounding the anode with a porous clay diaphragm. The anode is therefore immersed in electrolyte inside a porous clay cell or dish and the rubber particles together with those of zinc oxide, sulphur, etc., are deposited upon this porous so-called anode diaphragm. Thus the rubber as it collects forms a continuous, homogeneous, tough covering of uniform thickness. Any thickness up to an inch or more is practicable. Rubber thus formed is stronger than rubber prepared by the old methods, and is free from gas or air holes. Therefore it is permissible to use less thickness for some purposes.

Industrial development is already well advanced. Continuous, automatic production of certain kinds of articles is feasible. Manufacture of inner tubes for automobile tires is the most important application, if quantity be the criterion. Bathing caps, stationer's elastic bands, tobacco pouches and hot-water bottles are other examples. Insulation for wires and other things electrical is another application. After being formed of desired thickness, articles have to be dried, cured, and removed from the molds. The process may be operated as a continuous cycle. Drying and curing steps also differ advantageously from the old methods. No high temperatures are used. For impregnating textiles the rubber can be more intimately applied to the fibers.—W. C. Geer in *Research Narratives*, vol. 7, no. 4, April, 1927.

Use of Power in Steel Mills

IN THE middle of the last century the only power used at a blast furnace was for blowing. The filling of the furnace and the disposal of its products was all hand labor. At that time furnaces had a production of perhaps 25 to 50 tons per day. At the present time practically the only duty of the workman is to intelligently operate the mechanical devices provided to perform the various functions required. Today, with only a small fraction of the number of men that were previously employed, we produce 700 to 800 tons of pig iron from a single furnace, at a much lower cost, although wages have probably quadrupled, and supplies are carried hundreds of miles. The same applies to the steel works and mills, and probably the most important item in the development of the industry has been the ready availability of power, which can be substituted for human effort. European plants, however, are not so completely equipped with labor-saving devices, with the result that in a number of German plants, which can be considered typical, the production per man is only about 40 per cent of what we obtain.

About 35 years ago the first efforts were made to use electric motors in the steel industry, the first applications being in locations where other mechanical methods were unsatisfactory. After about 15 years the motor had practically supplanted other sources of power for auxiliary drive. In general it has been found that direct-current motors have more desirable characteristics than alternating-current machines. Rotary converters or motor-generator sets are usually distributed throughout the mills to supply the direct current. Today electric power is used for driving the main rolls in a very large proportion of the mills, and is used universally for auxiliary drives.

It is estimated that the total amount of power used in the industry in 1926 was equivalent to about 7,000,000,000 kw-hr., approximately 75 per cent of which was generated in the mills and 25 per cent purchased. The approximate rating of motors used in steel mills is 3,500,000 hp., and of this total somewhat over 1,000,000 hp. are driven by purchased power.

In the early days reciprocating-engine-driven alternators were used for power generation in the plants. About 20 years ago there was considerable interest in the development of gas-engine-driven units, the engines being supplied with blast-furnace gas. They were originally developed in Europe and were used quite extensively prior to any general adoption in this country, largely on account of the higher cost of fuel in European countries. After many initial difficulties gas engines were developed in the United States and gave satisfactory operation; however, they were costly and expensive to maintain. In the meantime the turbo-generator was being rapidly developed, and the introduction of successful turbine-driven blowers for blast furnaces has been a very important factor in determining the methods of generating power in steel mills.

With the development of central-station power plants in the mills the use of electric motors for all purposes became more desirable, and the facility with which electric power can be transmitted has been a material factor in affecting the layouts of modern plants. The growth of large stations and consequent reduction in cost of power has made it desirable for many plants to purchase power instead of generating it.

In steel mills with blast furnaces the power is usually produced in part from surplus blast-furnace gas. In making any plans to utilize surplus blast-furnace gas it is important to consider all factors that enter into the question, for many theoretical estimates have been based upon estimated quantities of surplus blast-furnace gas which in practice have been found too optimistic.

Modern furnaces require air volumes in excess of those which can be supplied by single gas-engine-driven blowers, and the latest development in this field is the turbo-blower. Its use in the United States has increased rapidly in the last few years, due to improved economy of later designs, and the possibility of building a single unit in any capacity that may be required in practice.

Owing to the fluctuation in the demand for power in a steel plant and the fact that there is little correlation between the operation of the blast-furnace plant and the power requirements, it is obvious that it is not an ideal load for an internal-combustion engine. To obtain better results it has become regular practice to install about 30 per cent of the total capacity of turbo-generators, which are used to carry the peak load. The efficiency of such a plant, however, is not very much in excess of that obtainable with modern turbo-generating plants, the two types of prime movers complicate matters, and the cost is much greater than an all-steam station.

It is estimated that if all the plants with blast furnaces were equipped with generating plants of high efficiency the surplus gas would be approximately sufficient to supply all their power needs. In plants with open-hearth furnaces discharging gases at 1200 to 1500 deg. fahr., it has been found that from 1200 to 1500 lb. of steam per ton of steel can be generated.

The power required for driving rolling mills varies with the type of mill and its product. In the largest units the motors driving the rolls may develop momentary outputs of 10,000 to 15,000 hp., with an average of 6000 to 7000 hp. The largest continually rated motor installed at present for this work is 9000 hp. Variable-speed motors being desirable, motor-generator sets are generally employed to supply direct current to permit easy control. Such drives are not quite as efficient as the various alternating-current combinations, however, but have the advantage of simplicity. Most of the auxiliary motors are of especially sturdy construction, and usually do not exceed 100 to 150 hp. each. The amount of power consumed by these auxiliaries is large, often amounting to one-third of the total.

Future practice will probably tend toward efficient steam-driven turbo blowers and turbo-generators. Boiler plants will burn blast-furnace gas economically, with provision for supplementary fuel, possibly pulverized coal in the same setting. There will be a tendency to interconnect these plants with central stations. In general it will be found that a steel plant is not justified in generating its own power unless fuel that otherwise would be wasted is available.—From paper on Use of Power in Steel Mills, by W. Sykes presented at Second Mid-West Power Conference, Chicago, Ill., February 15 to 18, 1927.

French Industrial Machinery Market During 1926

THE French industrial machinery market during 1926 was characterized by a new production record and by the maintenance of the high export level of the preceding year. The latest available statistics on the French production of industrial machinery are for the year 1913, and place the value of production for that year at 225,000,000 francs, or roughly $45,000,000. It is estimated by a number of prominent local manufacturers that the gold value of the production for 1925 was more than two and a half times the value of the 1913 volume, or approximately $116,865,000 (average value of franc in 1925, $0.0477). A recent estimate places the 1926 production volume somewhat in excess of the gold value for 1925.

The production of machine tools in 1913 amounted to 10,000,000 francs, or roughly $2,000,000. The production of machine tools during 1925 is estimated at more than three times the gold value of the 1913 volume, or approximately $7,000,000. Various estimates on the 1926 production place it slightly in excess of the gold value for 1925.

Because of the wide fluctuations in the value of the franc, comparisons are made here upon the basis of tonnage. Exports of industrial machinery rose to new record levels during 1926, amounting to 100,191 tons as compared to 77,396 tons in 1925. Imports, on the other hand, fell off to 72,440 tons, as compared with 76,734 tons in 1925. Imports of industrial machinery in 1913 totaled 126,294 tons, and exports 27,243 tons. A comparison of the 1913 and 1926 statistics shows that French imports fell off 53,855 tons, or 43 per cent, while exports increased 72,948 tons, or to 367 per cent of the earlier volume. Disregarding 1919, 1920, and 1921 (the abnormal post-war years), industrial machinery imports from the United States have increased steadily, and in 1925 amounted to 14,543 tons, or almost double the 1913 volume. Imports from Great Britain and Germany, on the other hand, showed losses in 1925 of 10,363 tons and 41,481 tons, respectively, from their 1913 trade.—*Commerce Reports*, May 23, 1927.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

Dopes and Detonation

In *Mechanical Engineering* of June, 1926 (p. 616), there appeared a brief abstract of the first part of an investigation by Prof. H. L. Callendar, Capt. R. O. King, and Flying Officer C. J. Sims on detonation phenomena in internal-combustion engines. A further and more complete instalment of a new investigation on the same subject follows.

The investigations forming the basis of this report were undertaken at the Air Ministry Laboratory, Imperial College of Science and Technology, London. The primary object of the investigation was the determination of the physical actions that delay or prevent detonation in an engine cylinder.

Effect of Compression Ratio

The effect of raising the compression ratio is invariably to increase the tendency of any susceptible fuel to detonate under given conditions, as would naturally be expected, since both pressure and temperature increase together when the compression ratio is raised. This point has been most carefully investigated by Ricardo, who finds that there is a practical limit to the compression ratio which can be employed (with any given fuel under conditions of maximum power) without producing excessive detonation, leading to preignition and loss of power if continued. This limit is called the "Highest Useful Compression Ratio" (H.U.C.R.) and is the highest compression ratio which it is worth while to employ with a given fuel.

There is no question at the present time that the H.U.C.R. is of primary importance in limiting the fuel economy obtainable with any fuel to which it applies, but there is still a good deal of controversy as to whether temperature or pressure is the most important factor in determining the onset of detonation. Many methods have been employed of raising the possible limit of compression ratio by the addition of other constituents to the mixture. The oldest and most familiar, consisting in the addition of a certain percentage of cooled exhaust gases, has been found to be one of the simplest and most effective in practice, and has the advantage for theoretical purposes that the resulting temperatures and pressures are amenable to calculation while the composition of the fuel itself is unaltered.

In 1521 Tizard made some calculations based on Ricardo's experiments in which H.U.C.R. was raised from 5 to 7 by this method with a given fuel. He came to the conclusion that the maximum flame temperature remained practically constant for different dilutions and was the determining factor in causing detonation.

Ricardo himself, on the other hand, has always been of the opinion that the pressure was the most important factor. On repeating the experiment in greater detail with various diluents (*Automobile Engineer*, June, 1924) he found that, whereas the flame temperatures and explosion pressures varied considerably at the detonation point with different degrees of dilution, the rise of pressure on ignition remained remarkably constant throughout the series. This seems to be a very reasonable result in view of the fact that ignition by compression must depend greatly on the rapidity and extent of the rise of pressure during ignition. But the effect of other additions to the fuel cannot be treated on the same lines, or expressed simply in terms of rise of pressure or temperature.

Nuclear Drops as Foci of Ignition

When gasoline is atomized in a current of air, the drops as they evaporate leave a residue or nucleus consisting chiefly of the less volatile constituents, such as the heavier paraffins. It seems to be well established, as previously explained, that the heaviest paraffins have the lowest ignition points, as compared with other common constituents of gasoline. The ignition point of these residual drops would tend to fall as evaporation proceeded, and would be lowest when they became evanescent. The drops at this stage may be called "nuclear drops" and are of microscopic size.

When a mixture containing such drops is being rapidly heated by compression, the nuclear drops are most likely to serve as foci of ignition, because they have a lower ignition temperature than the completely vaporized mixture. Moreover, it is a matter of common experience that chemical action (or adsorption, which usually proceeds such action) starts most readily at the surface of separation between liquid and vapor, because the surface molecules are in a state of intense strain due to surface tension. The ignition point of a mixture containing drops in suspension is thereby lowered. The drops would also act far more energetically than the vaporized mixture in absorbing radiation, which must be very intense at a late period of inflammation. This effect would be materially assisted by the carbon or smoke particles, which are always present in large numbers, and would naturally tend to become concentrated in the nuclear drops.

Since the nuclear drops form a relatively small percentage of the total fuel, it becomes easier to understand why very small quantities of suitable dopes should be capable of producing such striking effects on the tendency of a fuel to detonate. It is sufficient to suppose that the dope becomes concentrated chiefly in the drops as they evaporate, and infects them in such a way as to delay their ignition. The assumed concentration of the dope in the nuclear drops would follow naturally from the introduction of the dope as a constituent of the fuel. But further experiments were required to discover the various ways in which different dopes might act in raising the ignition temperature. With this object some typical dopes were selected and were tested in regard to their relevant physical properties at the temperatures and pressures occurring during compression in an engine, concurrently with corresponding tests of the same samples for anti-knock properties in the variable compression engine.

Three Types of Detonation

There appear to be three possible types of detonation: (1) The pressure-wave type, depending on the development of a pressure difference into a compression wave of sufficient intensity to fire the heated mixture; (2) the nuclear type, depending on the presence of nuclear drops or particles of relatively low ignition temperature; and (3) the molecular type, in which the temperature of a completely vaporized or gaseous mixture is sufficiently raised to induce direct molecular combination, even in the absence of pressure waves or nuclei. All three types are probably present to a greater or less extent in most cases, but any one may predominate over the others in imposing its own special character on the resulting detonation.

Conclusions

It is considered that the results of the experiments tend to support the nuclear theory of self-ignition.

It has been shown that the heavier paraffins, on account of their high critical temperatures combined with low critical pressures, are exceptionally liable to persist in the form of nuclear drops, which serve as foci to simultaneous ignition by compression owing to their low ignition temperature.

The marked effect of pressure in promoting detonation, established by the work of Ricardo and Fenning, is explained by the

rapid increase of nuclear condensation with increased density of the charge.

The action of a dope in delaying detonation is to infect the nuclear drops in such a way as to delay their ignition. The fact that these drops form a small percentage of the whole mixture helps to explain the possibility of a relatively small quantity of the dope being effective.

It has been shown that lead ethide and nickel carbonyl, two of the most effective metallic dopes, when mixed with gasoline residues, decompose rapidly at temperatures above 200 deg. cent., depositing a film of metal on the surface of the liquid. This metallic film would tend to protect the nuclear drops from oxidation, and would help to keep down their temperature by reflecting radiation.

Organic dopes, such as methylaniline and xylidine, have the advantage that much higher compression ratios can be employed than in the case of metallic dopes without risk of fouling the engine with deleterious deposits. On the other hand, much larger quantities are required than in the case of lead ethide. Organic dopes probably act mainly by the dilution of the nuclear drops, which results in a rise of ignition temperature; but the chemical reactions which may occur are very complicated and require further investigation. So little has been done in this direction that it is possible that some simpler and more effective organic dopes may be discovered when the chemical side of the problem has been adequately explored. (Prof. H. L. Callendar, assisted by Capt. R. O. King and Flying Officer C. J. Sims, *Aeronautical Research Committee, Reports and Memoranda, No. 1013* (E. 18), November, 1925, pp. 1–54, 15 figs., 6 tables, *etA*)

Short Abstracts of the Month

AERONAUTICS

Boundary Layer and Improved Performance of Aircraft

Wind-tunnel tests have shown that wings can be made with a lift-drag ratio of considerably more than twenty, and that streamline bodies may have a resistance which at high speed (say, 160 m.p.h.) is so small for a body of a cross-section of a modern transport machine that the lift-drag ratio of the whole could still be over twenty. The problem of building efficient airplanes is not, however, quite as simple as that. The complications are due to the presence of a landing gear, engine with its cooling, exhaust and induction piping, pilot location, machine guns in military planes, etc. All these factors make it impossible to use the ideal forms and force the designer to "break the lines" at many places. To appreciate the effect of disturbances of that character it becomes necessary to know in what way they alter the air flow along the bodies. The knowledge of this subject has been developed in the last years to a considerable extent, and forms what is known as the theory of the boundary layer, which, among other things, attempts to explain the conditions which lead to the phenomenon of "departure." This occurs when the general flow departs from the surface of the body, and is accompanied by the production of a "wake" or a region filled with eddies. The conditions of departure are of interest, because the departure of the flow from the walls of a body has an important influence on the whole flow and therefore on the efficiency of an airplane part. From this theory, originated by Prandtl in 1904 and presented in a more complete form in the original article, it would appear that: (*a*) The thickness of the boundary layer is small in comparison with the radius of curvature of the surface; (*b*) in the boundary layer the pressures are the same as those in the outer flow (the pressure gradient normal to the wall may be neglected); and (*c*) at the wall the speed is zero. The movement of a part of the boundary layer depends apparently on the resultant of the following components: (*a*) friction with the wall; (*b*) pressure gradient along the wall; (*c*) inertia forces.

If the pressure gradient is directed with the principal flow, the particles of the boundary layer will be accelerated. If on the contrary the pressure increases when moving with the principal flow, the boundary layer will be decelerated, thus coming to rest or even showing a contrary movement. In this case the pressure gradient is directed against the movement of the outlet flow. The author calls this an "adverse pressure gradient."

Fig. 1 shows what happens when the movement of a part of the boundary layer is reversed.

It gives rise to a stagnation, and the layer departs from the surface. An unstable sheet of discontinuity is formed and is rolled up, giving rise to eddies. Under certain limiting conditions (with certain restrictions) the flow pattern has been calculated for a laminar boundary layer in an adverse pressure gradient.

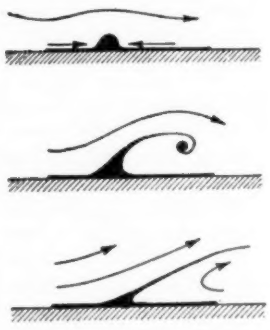

Fig. 1 Reversed Motion in the Boundary Layer Produces Departure

Several means have been considered for preventing departure, and the author discusses them in some detail. In wings the lift reaches a maximum as the angle of incidence increases. This is due to the departure of the flow from the upper surface and is called "stalling" in the case of aerofoils. A figure given in the original article shows the pressure on one such wing (R. A. F. 15). The general character is the same as of other simple sections. The pressures on the lower surface are plotted at the top. The maximum is near to the landing edge, decreasing to the rear.

Thus the boundary layer is accelerated by the pressure drop, in the direction of the main flow. This explains why disturbances on the pressure side have only a small influence.

The contrary is seen on the upper side; the depression is shown in the lower part of the figure to have its maximum near the leading edge. Behind this point there is an appreciable adverse pressure gradient.

When the angle of incidence increases the pressure gradient increases till a maximum is reached and departure is developed (stalling).

Consequently the defection of the flow is not increased when the angle is still further increased; and there is a maximum value of the lift.

By fairing the leading edge of the aerofoil the pressure curve is smoothed, showing a less sharp peak. This explains why thicker aerofoils and those with a blunter leading edge show a higher maximum lift coefficient.

When aerofoils are disturbed at the upper surface, especially near to the leading edge, the maximum lift is decreased. It has been shown that a disturbance produced only in the center will give an appreciable decrease in lift and increase in the drag. This drag results partly from the energy dissipated by the irregular eddies formed, but also from the increase in induced drag. The aerofoil has no longer a lift distribution which resembles the elliptical one, but instead the lift grading curve is split up into two parts. Both of these have an increased downwash as they are comparable with two separate aerofoils of about half the original aspect ratio.

Among means by which the lift is increased and the drag reduced the author mentions slotted wings, with which tests which showed the effectiveness of this device were made by Betz, who had in mind the theory of the boundary layer. Another means of preventing departure has been only recently made known, but was experimented with by Prandtl as early as 1905. This is sucking fluid into the body.

A more recent application of this phenomenon has been made by Betz and Ackeret. The upper surface of a model aerofoil is partially made of wire gauze through which air is sucked into the model. Departure now only occurs at a much greater angle of incidence, with a greater lift. The provisional tests indicate that a lift coefficient of 2 (Br. abs. units) may be attained. The energy necessary for sucking the air into the wing may be expressed as a drag coefficient, and this was found to be smaller than the profile drag of the aerofoil.

The resistance of a body may be decreased by diminishing the area at which departure is produced. Experiments of this sort have also been made at Göttingen with a sphere the rear half of which was made with wire-gauze strips. The resistance of the

sphere could be reduced at 20 per cent of the value for a normal sphere under the same conditions.

Wonderful applications of these methods seem possible. The deflection of an air current through 180 deg. was produced by sucking air into a semi-cylindrical wall of wire gauze.

Fig. 2 shows a diagram of the experiment. When no air is sucked in, the air current is directed as indicated by the arrows in the upper sketch. In the lower sketch we see that with suction the arrows indicate a complete deflection.

Conclusions. For practical work we may conclude: Even

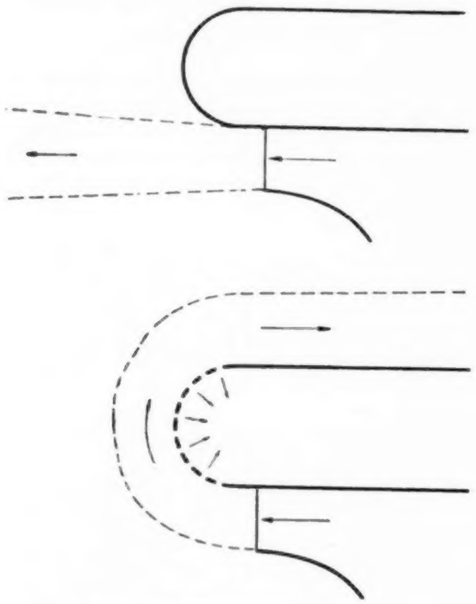

FIG. 2 JET OF AIR, WITH AND WITHOUT SUCTION

small disturbances should be avoided especially on the suction side of wings. Slotted wings and rotors show how departure may be prevented. The new method of sucking air into the body may be applied in several ways. The author doubts whether applications on the whole length of a wing will be practicable. He imagines that this method will be useful for diminishing the damage done by struts, etc., to bodies or wings, by sucking air into the main part over a small area behind the inevitable disturbance. The quantity of air which is necessary to suck in is relatively small, e.g., smaller than the intake of the engine. (Paper read by A. G. von Baumhauer, Sub-Director of the Government Aeronautical Laboratories, Amsterdam, under the title, Some Notes on the Possibilities of Progress in Aviation, before the Institution of Aeronautical Engineers, published in *Journal of the Institution of Aeronautical Engineers, Great Britain*, vol. 1, no. 4, Apr., 1927, original paper, pp. 5–25, discussion 25–29, 20 figs., *etA*)

Ford Air-Transportation Data

AIR-MAIL service from Detroit to Cleveland and Chicago conducted by the Ford Motor Company achieved its first birthday on February 15, 1927. During the year ending on that day a total of 8301 lb. and 4 oz. of mail was carried by Ford planes. During the same period the same planes carried 1,807,116 lb. of company freight to Cleveland and Chicago, or for forwarding from these points. This freight is the equivalent of about 72 minimum freight carloads. During the year, 600 flights between Detroit and Chicago were scheduled. Five hundred and eighty of these, 96.6 per cent, were completed. During the same period 979 flights between Detroit and Cleveland were scheduled, of which 963 or 98.3 per cent were completed. The year's flying, 138,320 miles in the Chicago service and 122,301 miles in the Cleveland service, brought the total Ford line mileage since April 13, 1925, to the impressive figure of 453,999 miles. Since the beginning of scheduled flights, 2459 have been completed or 97.3 per cent of those scheduled. (*The Ford News*, Mar. 22, 1927, p. 1. Abstracted through *Automotive Abstracts*, vol. 5, no. 4, Apr. 20, 1927, pp. 100–101, *g*)

AIR ENGINEERING

Multi-Stage Air Compressors

THE efficiency and the reliability of the motorship depend to a large extent upon the means adopted for obtaining and maintaining a sufficient supply of compressed air for starting and maneuvering purposes and—unless airless fuel injection is employed—for atomizing and injecting the fuel into the cylinders. It has become almost standardized practice to compress the air to a working pressure of about 1000 lb. per sq. in. for fuel-injection purposes, by passing it at a lower pressure—usually about 500 lb. per sq. in.—to the starting-air tanks. The production of compressed air at 1000 lb. per sq. in. demands the use of a multi-stage compressor, and it is a matter of some difficulty to determine exactly what is the correct number of stages to employ. Owing to the rise of temperature which accompanies the increase of pressure, it is essential that coolers shall be fitted between the various stages of the compressor, and these, of course, with their water connections, tend to add to the complication of the unit as a whole. It is found, however, that unless this cooling of the air between stages is efficiently carried out, the temperature of the air rises to such a figure that carbonization of the lubricating oil may occur, with the result that trouble will be experienced with piston rings and liners, while the temperature stresses in the various compressor cylinders may also lead to failure. Even if efficient intercooling is arranged for, it is desirable that the rise of temperature in the cylinder during compression should not become excessive, and this points to the desirability of using three or even four stages of compression in preference to only two. Experience with both types has shown that the three-stage compressor is far more reliable than the two-stage, and this reliability extends even to the blast-air bottles themselves. Furthermore, the three-stage compressor has the advantage of requiring rather less power for driving, other things being equal, than the two-stage machine, while the four-stage compressor represents an improvement on the three-stage type on this latter point. On the whole, however, the three-stage machine is probably the best compromise (of all types) for marine use. (*The Marine Engineer and Motorship Builder*, vol. 50, no. 596, Apr., 1927, pp. 121, *p*)

ENGINEERING MATERIALS (See Power-Plant Engineering: Erosion Caused by Jets)

FUELS AND FIRING

Low-Temperature Carbonization in Vertical Retorts

THIS paper gives a preliminary account of the construction and behavior of the latest vertical cast-iron retorts erected at the Fuel Research Station of the Department of Scientific and Industrial Research in England for the low-temperature carbonization of bituminous coal. It gives a description of the retorts first built, which proved to be too narrow to allow of the continuous passage of small coal if of caking or medium-caking quality. The two wider retorts next built have been in use for twelve months and are 21 ft. high × 6 ft. 6 in. × 7 in., widening to 6 ft. 10 in. × 11 in. They are made of ordinary gray cast iron of good quality. The extractor gear is described in some detail.

Use was made of the experience gained in connection with heating the narrow retorts when the wider retorts were completed. The heating is arranged on a system which at the Fuel Research Station has been found particularly successful: namely, the fuel gas is led by means of plain steel pipes bent vertically upward at the end into various corners built into the internal face of the setting. It was found that the chimney effect of the corner enables the flame and hot gases to cling closely into the corner and not to wander, and so contact with the retort walls is avoided. The heat from the flame is conducted along the brickwork, so that the retorts are heated very largely by radiation from the flame and brickwork. It appears certain also that the waste gases must heat the retorts by convection during circulation inside the setting before leaving the chamber after their first passage direct to the top. There are three sets of this type of burner, dividing the vertical height of the retorts into three portions.

As regards the coal used, at first an effort was made to handle only non-caking slacks. Whenever the available supplies became short on account of the coal strike all kinds of coal were used, among others strongly caking varieties, and all have been worked through the retorts quite satisfactorily. It was noticeable, however, that the ease of working, output, and yield of tar increased in proportion to the length of time that the coal remained in lump form in the retort.

Thus, non-caking nuts were the easiest to work, gave the highest yield of tar, and would give the greatest throughput. Slightly caking nuts would probably approach this very closely. Strongly caking nuts required no rodding and gave a good throughput and a good yield of tar. With a mixture of lump coal and fines some rodding was necessary, and the rate of throughput and the yield of tar depended on the proportion of fines. With fine coal both the throughput and yield of tar were low, and rodding amounting to possibly eight minutes per two hours was necessary on each occasion of charging.

It was found that the fine coal (through a $^3/_8$-in. screen), which amounted to 50 to 60 per cent of the Durham coal used, could be dealt with by briquetting. Several tons of ovoid briquets were made with 4 to 6 per cent of pitch as binder, and it was found that these briquets, when mixed with the lump coal, could be carbonized at an increased throughput. When, however, the briquets were passed through the retorts by themselves, the coke formed was so hard and strong as to be difficult to extract. The small coal was therefore mixed with varying percentages of breeze prior to briquetting. Of these mixtures, 20 per cent low-temperature coke breeze, 74 per cent coal, and 6 per cent pitch was the most satisfactory. These briquets passed through the retort at a good throughput and formed excellent coke.

The coke from the fine Durham coal was very strong, hard, and compact, and arrangements have been made to test its suitability for foundry purposes.

The amount of fuel gas required may be judged by the following figures for operation of run-of-mine Durham coal without steam, at the rate of 2.7 tons per retort per day:

	Water gas per hour, cu. ft.
Lower set of burners....	1050
Middle set of burners...	420
Upper set of burners....	1000

The calorific value of the gas varied from a maximum of 752 B.t.u. per cu. ft. for preheated nut coal to as low as 547 B.t.u. per cu. ft. It is considered, however, that the calorific value of the gas can be improved as the unusually low values shown in some cases are said to be due to admixture with air which leaked through the joints of temporary retort covers. (C. H. Lander, Director of Fuel Research, and J. Fraser Shaw, Ch. Engr., Fuel Research Station, Department of Scientific and Industrial Research of Great Britain, Fuel Research, in Technical Paper no. 17, 8 pp. of text, 3 pp. of tables and illustr., *e*)

Oil versus Coal

At the jubilee meeting of the Cunard Steamship Company, Ltd., which took place last week at Liverpool, Sir Thomas Royden, chairman of the company, made an interesting reference to the relative costs and operating expenses of oil- and coal-burning ships. Most of the large Cunard liners are oil-burning vessels, and it is not surprising to learn that the oil-fuel bill of that company is actually larger than the bill for fuel which is paid by the British Admiralty. After enumerating the advantages of oil, which are briefly summed up in the quick turn around obtained, the general cleanliness of the fuel, and the reduced engine-room staff, Sir Thomas went on to say that against these great advantages the unfavorable cost of fuel oil compared with that of coal had to be considered. There were limits, he thought, to the price which a shipping company could afford to pay for oil fuel, and it was quite possible, he continued, that if the price of oil continued to rise the Cunard Company might be required to give serious consideration to a reversion to the use of coal, at any rate for a part of its fleet. There is among shipowners and marine engineers an increasing opinion that if oil fuel is to be made use of aboard ship it should be reserved for burning directly in the cylinders of oil engines or for raising steam in high-pressure boilers which would serve efficient geared turbine machinery. It is not unlikely that renewed attention will be given to the claims of coal in the near future, and the results obtained from mechanical stokers fitted to marine boilers and from powdered-fuel plants for ship use will be awaited with interest. (*The Engineer*, vol. 143, no. 3817, April 15, 1927, p. 401, *g*)

Sauvageot Grate Firing

This type of firing is said to have been found particularly adaptable to the use with gas producers. It consists essentially of a grate of peculiar construction, an air box, and mechanical drive. The grate proper consists of grate bars or rollers, such as shown in Fig. 3, which are cast-iron hollow rollers with a large number of

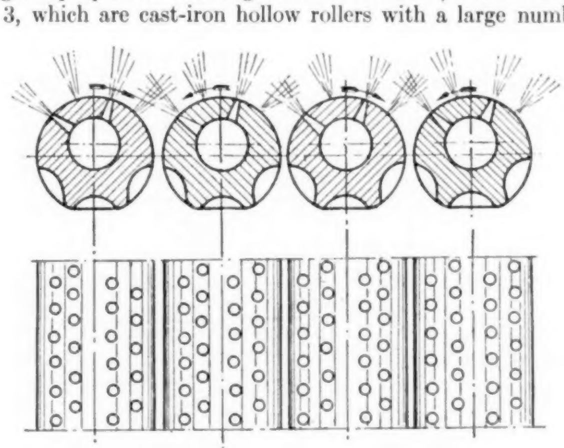

Fig. 3 Grate Bars of the Sauvageot Grate

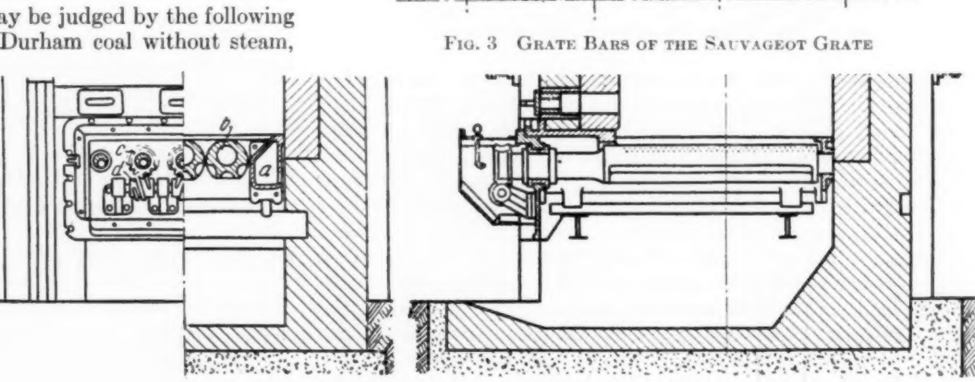

Fig. 4 Gas Producer with Sauvageot Grate

nozzles of conical shape contracting outward. The part of the surface not occupied by the nozzles is shaped so as to break up and forward the fuel. For this purpose the rollers are equipped with ribs and depressions indicated in Fig. 3. The rollers are provided at both ends with journal bearings equipped at the forward end with gear drive. The gears *c*, Fig. 4, engage with the worm *d*, cut with the thread alternating to the right and left so as to drive in opposite directions contiguous grate bars. The worm may be driven either by hand or from a source of power.

The air box contains the shaft driving the worm and also a pipe through which steam or water is delivered to the combustion air. From the same pipe, branches equipped with valves are led to each of the grate rollers.

The air is led under pressure into the grate rollers and discharges through the nozzles into the combustion chamber. Because of the arrangement and shape of the nozzles the air expands on entering the combustion chamber, which together with the large number of nozzles produces extremely fine atomization of the combustion air over the entire surface of the grate. It is claimed that because of this and also because the air is injected sideways an unusually good combustion is secured, this being materially assisted by the method of introducing the steam. This is said to be vigorously picked up by the air from the branches of the main steam pipe and delivered

to the combustion chamber in a finely atomized form. The valves located in the branch pipes permit regulating the admission of steam in such a manner that it can be delivered where it does most good, particularly where slag clinkering is especially bad, and interferes with proper distribution of the air flow.

Tests made by Professor De la Condamine, Director of the Experimental Laboratories of the Office of Rational Heating in Paris, have indicated a saving in fuel ranging from 20 to 40 per cent as compared with other methods of combustion.

One advantage claimed for the new system is that its tightness is secured largely by the use of the air of combustion itself where in other grate systems a water seal is used. The latter has the disadvantage that the water taken up by the hygroscopic slag is split up or evaporated in the combustion space and takes up heat uselessly. It is also claimed that because of the fine atomization of the air and possibility of exactly controlling the admission of steam, unnecessary disturbance of the fire zones is avoided, which permits a more uniform combustion and hence more uniform gas generation. The possibility of closely controlling the steam admission is of particular interest for operations where it is important to have the gas as clean as possible, such as, for example, glass works.

Tests made by Professor De la Condamine dealt with a comparison between a Siemens gas producer with an ordinary grate and a similar producer equipped with a Sauvageot grate. As fuel in both cases was used of the easily coking Saar coal and with admission of some coke. In the Sauvageot producer in 6 hours 470 kg. of coke and 336 kg. of coal, or a total of 806 kg., were burned, equivalent to 86 kg. per sq. m. of grate area per hour. The depth of the fuel bed was on an average 1.20 m. (47.2 in.). The complete analysis of the gases produced in both cases is given. The content of the carbon monoxide in the producer equipped with standard grates was slightly in excess of 28 per cent, in the one with Sauvageot grate better than 32 per cent, the lower heat values being respectively 1032 and 1263 large calories. It is stated that the Sauvageot grate was installed in France in more than 140 gas producers and is now finding a market in Germany. (Hans Lösche in *Die Wärme*, vol. 50, no. 12, Mar. 25, 1927, pp. 219–221, 5 figs., *d*)

The K.S.G. Process of Low-Temperature Carbonization

DESCRIPTION of the Kohlenscheidungs-Gesellschaft or "K.S.G." process of low-temperature carbonization, which has been in operation on a commercial scale since 1924 at the Matthias Stinnes I.-II. Colliery, Karnap, near Essen. The process was developed under the direction of the late Hugo Stinnes, and has since been acquired by the International Combustion Engineering Corporation of New York.

It is well known that the latter have just come to a working arrangement with the Imperial Chemical Industries, Ltd., formed by the amalgamation of Brunner Mond & Co., Ltd., Nobel's Explosives, Ltd., the United Alkali Co., Ltd., and British Dyes, Ltd., and having a capital of £56,000,000, and one of the chief objects is to be, it is understood, the development of improved methods for the utilization of raw coal. Coal Oil Extraction, Ltd., of London, associated with International Combustion Engineering Corporation have control of the McEwen-Runge process for the low-temperature carbonization of pulverized coal, now in commercial operation at the Lakeside Power Station, Milwaukee, with a 200-tons-per-24-hours retort, and one of the main results also of the above important association of interests will be the installation in Great Britain of a McEwen-Runge plant which, it is stated, will be capable of carbonizing 500 tons of raw coal per 24 hours.

Briefly, the process consists in the use of a long rotary, mechanically continuous, slightly inclined, cylindrical retort which is externally heated. The construction is that of one cylinder within another, the crushed bituminous coal or other raw fuel entering at the lower end, traveling up through the inner cylinder, falling through perforations at the termination of this travel, and then being conveyed back again along in the outer cylinder, which latter represents the maximum heating zone. That is to say, the raw coal is charged and the solid residual low-temperature fuel discharged from the same end of the setting.

The installation at the Karnap Colliery is one retort with a normal throughput of 60–80 tons of bituminous coal per 24 hr., with an overload up to 100 tons, and the length of the outer cylinder is 76 ft., while the diameter is 10 ft. The inner cylinder has a diameter of 5 ft. 8 in., and the speed of the retort is about one revolution in 90 sec., the total duration of the travel of the charge through both cylinders being approximately $2^1/_2$ hr., while the temperature at the hottest portion is 1112–1292 deg. fahr. (600–700 deg. cent.).

The most remarkable claim in connection with the Kohlenscheidungs-Gesellschaft process is that extremely hard, smokeless, low-temperature fuel, containing, say, 10 to 12 per cent volatile matter, can be obtained direct from a large, mechanically continuous retort of this description, having an output of nearly three tons of low-temperature fuel per hour, without any necessity either for briquetting or otherwise preheating the charge before carbonization, or using small intermittent retorts in which the charge is subjected to internal compression because of the expansion of the swelling and plastic material. This result is achieved by the sudden and rapid heating of the charge at one stage, combined with the action of a supply of highly superheated steam, which apparently distils sufficient of the volatile matter with such speed as to prevent any undue swelling.

The temperature of preheating in the inner cylinder probably does not much exceed 392–572 deg. fahr. (200–300 deg. cent.) during, say, $1^1/_4$ hr. travel, but the material then suddenly drops out of the inner cylinder into the top portion of the outer cylinder, which is arranged to be the maximum heat zone, that is, as already stated, 1112–1292 deg. fahr. (600–700 deg. cent.) in the combustion chamber, probably corrresponding on the average to 932–1022 deg. fahr. (600–700 deg. cent.) in the combustion chamber, and on the average to 932–1022 deg. fahr. (500–530 deg. cent.) in the actual charge itself. At the same time the superheated steam is blown in at this point and the combined effect of which—together with the rapid heat rise—prevents undue swelling. The remainder of the carbonization is then carried out in the outer cylinder with a gradual decline in temperature, the figure at the bottom, just before the actual discharge of the material, being 932–1022 deg. fahr. (500–530 deg. cent.) in the combustion chamber. (David Brownlie in *The Gas Engineer*, vol. 18, no. 611, March, 1927, pp. 59–61 and 74, *d*)

INTERNAL-COMBUSTION ENGINEERING (See also Air Engineering: Multi-Stage Air Compressors; Railroad Engineering: The Kitson-Still Locomotive)

Exhaust-Gas Heat Recovery

AT A MEETING of the Institute of Marine Engineers, held on Tuesday evening, April 12, Thomas Clarkson read a paper entitled The Recovery and Utilization of Heat from the Exhaust Gases of Internal-Combustion Engines. The author pointed out that over 30 per cent of the heat in the fuel is carried away by the exhaust gases, and that probably more than that percentage of heat is lost in the jacket water. The late Professor Nicolson's experiments demonstrated the importance of gas velocity, and the scrubbing action between heating surfaces and gases emphasized by Professor Parry are referred to as providing a useful starting point for the practical engineer in designing an ideal plant for waste-heat recovery.

Boilers which have been used hitherto for waste-heat recovery have been of an ordinary type, generally multi-tubular, or sometimes boilers of the Scotch type. It is hardly surprising, therefore, that the science of waste-heat recovery has not made more progress, as it is impossible to obtain the most efficient results from such boilers. The ideal boiler for waste-heat recovery should not only fulfil the conditions of impingement of gases on the heating surface, but it should also be entirely immune from expansion troubles. It is important also that the boiler should make an efficient silencer, and render it unnecessary to fit any other silencer. When the exhaust gases are not available, as, for example, when a vessel is in port, the boiler should be able to be conveniently heated by an independent oil burner. A waste-heat boiler should be of the most robust and foolproof type. It should work well with the minimum amount of attention, and it should have the lowest cost for maintenance.

A boiler for which it is claimed that it fulfils these ideal con-

ditions is the thimble-tube type, which was originally designed to meet the extremely stringent conditions met with in the problem of road transport. The boiler having demonstrated its remarkable properties in this service, it was suggested as the ideal type of boiler for waste-heat recovery from the exhaust gases of internal-combustion engines. Careful tests were carried out four years ago with a Vickers-Petters engine of the two-stroke type, 84–90 b.hp. These tests gave very interesting results, and at the same time indicated the necessity for suitable provision being made for decarbonizing the heating surface.

In the thimble-tube type of boiler, any deposit may be burnt off without straining the tube joints. The method also permits of a differential tube spacing, giving a high gas velocity through the boiler. The boiler may be run in a dry condition without disadvantage, owing to the gas temperature's never reaching a point sufficiently high to burn the tubes. The maximum gas temperature may be given as 700 deg. fahr. up to possibly 1000 deg. fahr. A usual figure for a four-stroke-cycle engine is 600 deg. to 650 deg. fahr., and in the neighborhood of 500 deg. fahr. for a two-stroke-cycle engine.

As a general rule, 300 deg. fahr. can be taken as the practical limit beyond which the gases should not be cooled, although there are cases where gases are required to be cooled below this temperature, and in tests conducted by Professor Cook at King's College, the gases were reduced in some cases to as low as 143 deg. fahr. The quantity Q of thermal units recoverable may be readily estimated by the following simple formula:

$$Q = \frac{\text{B.hp.} \times C \times D}{4}$$

C being a constant—in the case of four-stroke engines 12, and in the case of two-stroke engines 20; and D being the drop in temperature of the gases.

If hot water is to be produced, say, up to a temperature of 200 deg. fahr. the gases can easily be cooled another 100 deg., which, in the case of a two-stroke engine may represent an increase of 100 per cent in the heat recovered.

One difficulty which is met with by designers of waste-heat boilers is the unfortunate fact that there are at the present time no regular standard diameters of exhaust pipes in relation to horsepower. Some makers will use a 7-in. exhaust pipe for the same power of engine for which others will use a 10-in. or a 12-in. pipe; or, say, a 36-in. pipe in place of one of 24 in. From the point of view of heat recovery it is always desirable to keep the exhaust pipe as small as possible consistent, of course, with the avoidance of back pressure.

Back pressure has been regarded as a "bogey" by engine builders, more particularly in reference to engines of the two-stroke type, and it has been difficult to convince engine builders that the application of a properly designed waste-heat boiler need not create any back pressure, or in any way impair the thermal efficiency of the engine. It has been shown by careful tests that application of the thimble-tube boiler improves the volumetric efficiency without having any detrimental effect upon the thermal efficiency. In one case it was found that the fuel consumption for a given horsepower was reduced by as much as 3 per cent after a thimble-tube boiler was fitted to the exhaust pipe. Prof. Gilbert Cook, of King's College, University of London, has made careful tests—already referred to—on this point, and he has also shown that on the assumption that the temperature of feedwater can be taken as a basis, efficiencies as high as 86 per cent of heat recovery can be obtained. In this connection reference may be made to tests recently conducted on oil firing of thimble-tube boilers, in which efficiencies as high as $91^1/_2$ per cent were obtained.

With a properly designed waste-heat-recovery installation, it is possible to regenerate 5 per cent of the power of four-stroke engines by employing the best type of waste-heat boiler, in combination with a low-pressure turbine. This means that in the case of a vessel of 10,000 shaft hp., about 500 hp. can be generated from the waste gases, representing a saving of not less than two tons of oil per day, which would go far to provide all the power required for the auxiliary services of the vessel, including lighting, steering, ventilation, etc. (*The Engineer*, vol. 143, no. 3817, April 15, 1927, pp. 411–412, *ep*)

The Hill Diesel Engine

The Hill Diesel Engine Company, which has during the past six years built a line of engines from 6 to 30 hp. in one- to four-cylinder units, are now making 6-in. by 10-in. new four- and six-cylinder models, developing from 50 to 125 hp. at speeds of from 400 to 900 r.p.m.

The new Hill model employs a completely water-jacketed precombustion chamber, into which fuel is sprayed in a finely divided state. Here ignition occurs and combustion is carried far enough to increase the pressure above the compression pressure and to project the resulting gas into the cylinder with great force, where combustion is completed. A much-larger-sized precombustion chamber in proportion to the cylinder volume is used than the cups or chambers used in the early Hill engines, resulting in a much higher injection pressure for forcing gas into the cylinder. No effort is made in the new diagram to restrict the flow of gas from the chamber to the cylinder by means of small holes. Three $^5/_{32}$-in.-diameter holes are employed, and there is, therefore, no possibility of their plugging, regardless of overload or poor fuel.

A low-pressure fuel nozzle is fitted. This employs a spiral and large final orifice to break up the fuel, instead of the minute holes generally used. The complete fuel valve may be removed while the engine is running, which is a valuable point, and a substantial filter protects each nozzle from dirt.

Fuel injection takes place during the compression stroke, the time depending upon the nature of the fuel and the speed of the engine; and, with the above combination of precombustion chamber, spray nozzle, pump, regulator, and governor, all the functions are in full control of the operator over the entire range of speeds required in service. At full load the fuel consumption of the new engine is 0.48 lb. of 18,500 B.t.u. fuel per hp-hr., with an exhaust temperature of not more than 550 deg. fahr. (*Motorship*, vol. 12, no. 5, May, 1927, pp. 380 and 385, illustrated, *d*)

Light-Weight High-Speed European Oil Engines

The field for this type of engine appears at present to be for the operation of certain classes of rail motor cars, contractors' machinery, agricultural machinery, etc., where a comparatively

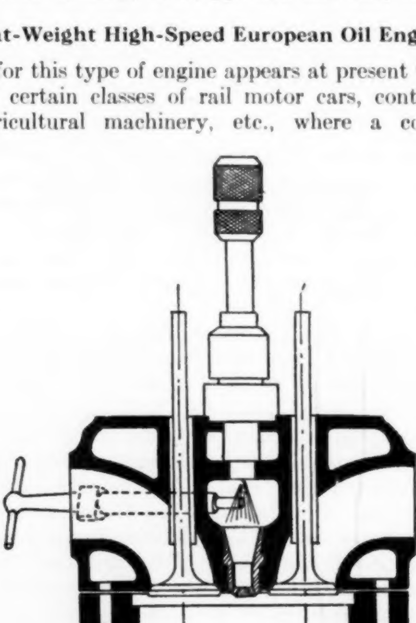

Fig. 5 Light-Weight High-Speed Benz Oil Engine

strong engine is required to operate continuously for long periods of time.

Under such circumstances the relative cost of fuel becomes a matter of importance in favor of the oil engine. In the marine field for the propulsion of light high-speed craft, this class of engine is also very well suited since there it has all the advantages of the gasoline type with the added one of safety and low fuel cost. The author describes five engines—the Beardmore, M.A.N., Benz, Bagnulo, and Mumford. Of these the M.A.N. is rated in the six-cylinder type to deliver 75 hp. at 250 r.p.m. There are two

injection nozzles per cylinder located at the side; the engine is of the solid-injection type with an open-nozzle type of spray valve and without any vaporization system. The fuel is supplied by independent variable-stroke fuel pumps. The Benz engine is rated at 25 hp. per cylinder when running at 800 r.p.m.; the cylinders are 5.3 × 7.75 in.; an aluminum casing forming the top half of the crank chamber has cast-iron cylinder liners pressed into it, while the heads are separate. The ignition chamber has the form of a spray bush at its lower end. Here are several small holes through which air on the compression stroke gains access to the combustion chamber. Here also the oil spray from the atomizer meeting with highly heated air becomes partially ignited. The pressure so created forces the explosive mixture into the cylinder where a relatively low pressure serves to ignite it. Each cylinder has its own independent oil pump of the cam-operated variable-stroke type which delivers the fuel at approximately 1000 lb. pressure to a spring-loaded type of atomizer. The milled head shown thereon belongs to a needle valve which can be opened at any time to bypass the fuel back to the supply and so cut out any individual cylinder. The component shown behind the left-hand valve is an auxiliary igniter used to facilitate starting. It consists merely of an arrangement to hold a roll of absorbent paper soaked in nitrate of potash, which is ignited with a match. The overhead rockers are protected by detachable covers. (F. Johnstone-Taylor in *Power Plant Engineering*, vol. 31, no. 9, May 1, 1927, pp. 514–516, *d*)

MACHINE SHOP

Knurling for Light Press Fits

Discussion of the application of knurled bearing surfaces in parts of small mechanisms. Knurling can be used to advantage in the manufacture of many classes of light mechanisms in which small stampings, pinions, or smaller parts must be assembled on their shafts and rods by pressing the parts together. The advantage of knurling in such cases is that it raises ridges or points and thus increases the shaft diameter sufficiently to allow for larger limits between the shaft and hole diameters than would be practicable if the parts were pressed together without knurling, i.e., with plain bearing surfaces. Furthermore, the knurling method of assembling light parts tends to avoid the cracking of bushings, hubs, etc., which might occur in assembling plain bearing surfaces, assuming that ordinary production limits are allowed. There are various kinds of knurled surfaces, the common styles being plain straight knurling and the diamond knurl. The former consists of knurls and ridges that are parallel to the axis, whereas in diamond knurling the grooves are at an angle and run crosswise from one side to the other. The plain or straight knurl is the kind employed ordinarily in connection with light press fits.

Knurls are made in a series of pitches, the pitch, according to common usage, representing the number of ridges or teeth per inch of circumference. The circular pitch on which the number per inch is based, is measured at the outside diameter of the knurl and in a plane perpendicular to the axis of the knurled part, as indicated in Fig. 6. The most common pitches are close to the pitches employed for standard screws. A table in the original article gives some of the more common pitches and the theoretical dimensions for circular pitches and tooth depths, as well as the minimum increase in diameter due to knurling. The application of this table is explained in connection with some examples.

When knurled surfaces are used for light press fits instead of plain surfaces, it is essential to know how much the diameter of the shaft, or whatever external part is to be knurled, will be increased by the knurling operation, in order to make proper allowance for the press fit. The theoretical dimensions for the depths of perfectly formed teeth should not be used in determining the increase in diameter of a knurled part, because in actual practice, teeth of full depth are not formed. Thus in producing a knurled part, say, on an automatic screw machine, the time for knurling may not be sufficient to form perfect teeth; moreover, the strain due to knurling causes some springing action, and consequently the actual depth of the teeth or knurls is considerably less than the theoretical dimension.

If 40 per cent of the theoretical depth is added to the shaft diameter, then the minimum increase in diameter due to knurling will be obtained close enough for practical purposes. Several examples of knurling for light press fits are given and methods for determining the pitches and tolerances indicated.

When a hardened pinion, collar, or other part is pressed on a shaft, a serrated hole in the external part is more effective than knurling on the shaft itself, especially if the shaft is not hardened. The teeth or serrations that are formed around the hole, as indicated by the diagram, Fig. 7, form grooves in the shaft when the two parts are assembled, thus providing a tight grip.

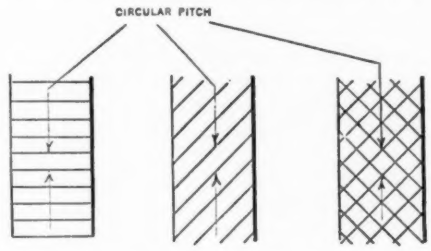

Fig. 6 Diagrams Showing How the Circular Pitch of a Knurl Is Measured

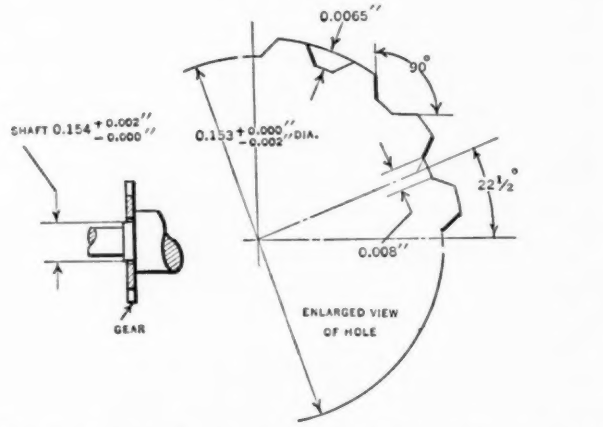

Fig. 7 Die-Cut Gear on Shaft, and Enlarged View of Serrations in Hole Which Provide Better Grip on Shaft

Fig. 7 shows at the left a small gear that has been assembled on its shaft in this manner, and at the right there is an enlarged view of the sixteen notches or serrations that have been cut in the hole of the gear. The gear teeth and these notches are blanked out in one separation. It will be noted that the teeth or serrations have float points. The press-fit allowance at the points of these teeth ordinarily varies from 0.001 to 0.005 in., a light fit usually being desirable, as an excessive allowance would damage the serrations; in fact, the full circumference of the hole is never used for the press fit. It is not always necessary to harden the gear, especially if the parts are small and not subject to rough handling or operation. (J. K. Olsen, Ch. Draftsman, Stewart-Warner Speedometer Corp., Chicago, Ill., in *Machinery*, vol. 33, no. 8, April, 1927, pp. 573–576, 7 figs., *p*)

MARINE ENGINEERING (See also Fuels and Firing: Oil versus Coal)

A Large Marine Oil Engine

Among the facts to which the author (designer of the Cammel-laird-Fullagar marine oil engine) calls attention, is that in 1922 only 15 per cent of the oil engines constructed were of the two-stroke type, whereas in 1926 this figure was almost 50 per cent. (The author does not state whether his figures apply to Great Britain alone or to world production.)

It is the author's claim that for equal power the two-stroke engine has fewer parts, and now that reliability, fuel, and lubricating-oil consumption are about the same, the choice of an engine will simply depend upon cost, weight, simplicity, and space occupied, in all of which the two-stroke-cycle engine is definitely su-

perior. As regards supercharging, the author believes that the means adopted are not without many drawbacks. Exhaust valves have always been a problem, and one would expect it to become more acute if mean pressures were deliberately raised to an extent sufficient to put the four-cycle engine on level terms in respect to special weight and cost with the two-stroke-cycle engine. (D. M. Shannon, British Manager of the Fiat Marine Oil Engine, in *The Marine Engineer and Motorship Builder*, vol. 50, no. 596, Apr., 1927, pp. 125–126, *g*)

The Bauer-Wach System of Combination Machinery

THIS represents a combination of reciprocating engine and exhaust-steam turbine so arranged that the advantages of combination machinery may be realized on single-screw cargo vessels of comparatively moderate powers.

The experimental unit consisted essentially of a triple-expansion engine having cylinders $13^3/_4$, $20^1/_2$, and $31^1/_2$ in. in diameter by $23^3/_4$ in. stroke and capable of developing about 450 to 500 hp. at 110 r.p.m. The turbine is mounted above the main-engine shafting and immediately abaft of the low-pressure cylinder of the reciprocating engine. The normal speed of the turbine is 600 r.p.m., and this is so reduced by means of double-reduction gearing that the turbine power is transmitted to the main shaft at 110 r.p.m. Maneuvering of the installation is effected by the reciprocating engine only. The power of the exhaust-steam turbine is transmitted to the main shaft through a Vulcan or Föttinger type hydraulic clutch coupling. By gradually filling the coupling after the reciprocating engine is under way the turbine is made to pick up its load without shock to the gearing. When it is desired to reverse the engine the coupling is emptied and the exhaust sent to the condenser by a bypass valve around the turbine. Owing to the considerable momentum possessed by the high-speed turbine, quick and easy means of isolating the turbine from the system during maneuvering is essential, and with this object the coupling and change-over valve are controlled by a common oil-pressure system, so that both may be operated automatically in proper sequence.

The performance data are given in the original article in the form of curves. With the turbine idle an output of 500 i.hp. at 110 r.p.m. was released with a condenser vacuum of 85 per cent. When the steam from the low-pressure cylinder was passed through the turbine the output of the combination unit was increased by 20 per cent, the vacuum being augmented by 7 per cent for this condition. When the vacuum was increased to 96 per cent the power increase was 33 per cent, all these results being obtained without any increase in total steam consumption.

The *Sirius* went into service at the beginning of September, since when the performance of her interesting propelling equipment has been very satisfactory. With a cut-off of 55 per cent in the high-pressure cylinder, the same sea speed has been attained as would normally be held with a high-pressure cut-off of 71 per cent, from which it has been estimated that the fuel saving due to the new type of machinery is of the order of 28 per cent. The steam is superheated to a final temperature of about 600 deg. fahr. when it leaves the boilers, the normal working pressure being 220 lb. per sq. in.

Since the completion of the *Sirius*, two fairly large steamers, the *Elberfeld* and *Ockenfels*, have been converted to the Bauer-Wach system. The first-mentioned vessel is owned by the Norddeutscher Lloyd and has a deadweight capacity of 9350 tons. Originally, her propelling machinery consisted of a single triple-expansion engine having cylinders $29^1/_4$ in., $47^1/_4$ in., and 28 in. in diameter by 54 in. stroke, superheated steam being supplied to the engines from coal-fired boilers at a pressure of about 215 lb. per sq. in. The original engines developed 3200 i.hp., and after the unit had been suitably modified and the exhaust turbine added, an aggregate power of 4000 i.hp. on the same fuel consumption was obtained. Trials in the loaded condition were carried out when the vessel was completed, and these showed that an increase of power of 29 per cent was obtained, after conversion, on the same fuel consumption.

The results of these trials are interesting as showing the results obtained with and without the exhaust-steam turbine in use at the same rate of revolution. The steam consumption of the installation in the first case was 15.34 tons of steam per hour for all purposes, and from this figure 2 tons per hour might be deducted for auxiliary consumption. The average output of the machinery was 2441 b.hp. and the condenser vacuum carried was 90 per cent. With the exhaust turbine in use and maintaining the same r.p.m., namely, 69, the combined b.hp. was 2363, but, on the other hand, the vacuum carried was 97 per cent, the feedwater temperature being the same in each case. Under these conditions the steam consumption was found to be 12.4 tons per hour for all purposes, or 10.4 tons per hour for the main engine only. A simple calculation will show that the relative steam consumption in the second case, on the higher-basis power of 2441 b.hp., is 10.7 tons, which indicates that a saving in steam consumption of rather more than 25 per cent was obtained with the Bauer-Wach exhaust turbine in circuit at a given speed of rotation of the propeller. The second vessel mentioned, the *Ockenfels*, is owned by the Hansa Line of Bremen, and has a deadweight capacity of 11,355 tons. This conversion was completed toward the end of the last year, and since the vessel was again placed in service her performance has been very satisfactory. (*Shipbuilding and Shipping Record*, vol. 29, no. 11, Mar. 17, 1927, pp. 301–303, *d*)

METALLURGY

Molybdenum in Nickel-Chromium Medium-Carbon Steels

THE addition of 0.6 per cent of molybdenum to carbon steels resulted in a marked rise of elastic limit, yield point, maximum load, and notched-bar impact figures, while the elongation remained satisfactory. There were, however, indications of serious segregations in steel containing 1.23 per cent of molybdenum, and the results obtained with this steel were irregular. With nickel steels, except for very high tensile strength, little further improvement results from additions of molybdenum above 0.6 per cent. A steel containing 4.5 per cent of nickel with 0.5 per cent molybdenum is only suitable when high tensile strength is required. Marked improvement in the properties of steels containing approximately 1.0, 1.5, and 2.0 per cent, respectively, of chromium resulted from the addition of 0.5 per cent of molybdenum. In the case of the 1.0 per cent chromium steel the properties were further improved by the addition of molybdenum up to 1.0 per cent, but no advantage was gained by increasing the molybdenum content beyond that amount. Additions of chromium up to 1.5 per cent to steel containing 0.5 per cent of molybdenum produced noticeable improvement.

Metallurgically, additions of molybdenum produce certain changes described in the original paper. Molybdenum has also a more powerful effect than nickel or chromium in reducing liability to imperfect hardening when the rate of cooling is slow. It also reduces the softening effect of tempering.

Nickel-chromium-molybdenum steels provide the best all-round combination of properties, but they are closely approached by certain nickel-molybdenum and chromium-molybdenum steels, each of which types shows some advantages and some disadvantages. For example, the chromium-molybdenum steels are distinguished by a high notched-bar-impact figure, though the elastic limit and yield point are slightly lower than those of nickel-chromium-molybdenum steels of the same hardness. Nickel-molybdenum steels, on the other hand, show higher yield ratios with considerably lower impact figures than nickel-chromium-molybdenum steels. Moreover, they are more difficult to machine when the nickel content exceeds 3 per cent, which it must do to enable a tensile strength of 60 tons per sq. in. to be obtained, and tests made after different rates of cooling indicate that they are more susceptible to mass effect than the nickel-chromium-molybdenum steels. (Paper by J. A. Jones of the Research Dept., Woolwich, published as R. D. Report No. 67; abstracted through *The Chemical Age*, vol. 16, no. 401, Mar. 5, 1927, pp. 17–19, *ep*)

The Carsil Chrome-Nickel Steel Process

THIS is an English process and deals with the manufacture of a high-grade chrome-nickel steel from Java iron sand. Some data as to the character of the product are given and preliminary cost figures cited. The physical properties of the material appear to be very satisfactory. The process is operated in England by the

Anglo-American Steel Co., Ltd., 66 Victoria St., London, S.W.1. (*Foundry Trade Journal*, vol. 35, no. 552, Mar. 17, 1927, p. 235, *d*)

MOTOR-CAR ENGINEERING

The Abramson Gearless Differential

DESCRIPTION of a differential which has been extensively tried out by the Skoda Works of Czechoslovakia, and is reported to have been adopted by this concern for use on the cars which it manufactures.

The principle can be gathered from Fig. 8, which shows part of a back wheel with hub and axle shaft in section. The axle shaft, instead of being in two sections, as with the ordinary differential gear, is in one piece and can be driven by a propeller shaft in the normal manner. The wheels are mounted freely on the axle, on the ends of which are fixed driving members *A*, which engage with driving stops *B* fitted inside the wheel hubs. When the vehicle is running straight the two driving members *A* are in contact with the driving stops *B*, so that the two wheels are rotated in the direction of the arrow at equal speeds. On rounding a curve the outer wheel will revolve more quickly on the shaft than the inner wheel, with the result that the driving stop *B* on the outer wheel will move away from its driving member *A* in the direction of the arrow. If the vehicle is then driven straight again the driving member *A* will gradually overtake the stop *B*, when the speeds of the two wheels will again be equal. Presumably this means tire slip. Similar action occurs, of course, when the vehicle is reversed.

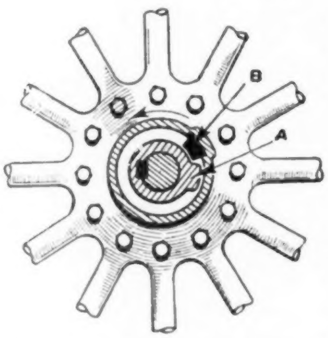

FIG. 8 THE ABRAMSON GEARLESS DIFFERENTIAL

Thus while the proportionate drive of the ordinary differential is lost and only one wheel drives round curves, the possibility of one wheel slipping indefinitely and leaving the drive, so to speak, "up in the air," is restricted to about three-quarters of a wheel revolution.

When the car is being driven in a straight course the differential is inactive and becomes active only when curves are described. With this construction the two sets of the differential mutually lock each other when one runs ahead on a curve, but the freedom of motion of the two wheels is not closely limited during straight-ahead motion. Several designs are described and illustrated, and a claim is made by the writer of the article that the new differential is cheaper to manufacture than the usual conventional type. (*Automotive Industries*, vol. 56, no. 15, April 16, 1927, pp. 573–574, 4 figs., and *Motor Transport*, vol. 44, no. 1151, Apr. 4, 1927, p. 415, *d*)

The Supercharged Straight-Eight Bugatti

THERE are indications, in France at any rate, that the supercharger hitherto confined practically to straight cars is to come into more general use. Recently the Bugatti factory put out a straight-eight sport-type four-seater roadster equipped with a supercharger and designed so as to be an intermediary between a racing and a pleasure car. No attempt has been made to bring the body out beyond the frame members. There are no running boards, two doors on the left side only, and wells are provided for the passengers' feet. Protection is given by a fixed wind screen, a very light folding hood is provided, and there is provision for baggage in the pointed rear. The compressor is of the Roots' type, mounted horizontally on the right side of the engine and driven from the timing gear at engine speed by means of a horizontal shaft with fabric couplings. The induction piping consists of a vertical branch flanged to the compressor and two horizontal arms, each one feeding four cylinders. There is a relief valve in the vertical arm to prevent damage to the blades in case of a blow-back.

It is claimed that with the special gear the car can attain 125 m.p.h. on the level, while with the standard gear it will run up to 112 m.p.h. The engine runs up to 6000 r.p.m. Most of the test runs were made on greasy winding lanes at the foot of the Vosges Mountains, and even under such conditions and running at 80 m.p.h. there was no tendency to slide on greasy turns. The fuel consumption is stated to be at the rate of between 17 and 18 miles per gal. (*The Autocar*, vol. 58, no. 1637, Mar. 18, 1927, pp. 423–424, 4 figs., *d*)

POWER-PLANT ENGINEERING (See also Internal-Combustion Engineering: Exhaust-Gas Heat Recovery)

Recent Developments in Zeolite Water Softening

THE article gives a history of the development of zeolite methods of water softening, beginning with the early synthetic zeolites and proceeding through the Bentonitic clay to the New Jersey green sand or glauconite as a softening medium and the precipitated type of zeolite. The most recent development in zeolites is a type of synthetic zeolite which the author designates as the "gel" type.

If solutions of sodium silicate and aluminum sulphate or of sodium silicate and sodium aluminate, of proper concentration are mixed in the proper proportions and under the proper conditions, no precipitate is formed. Instead, the whole mass gradually sets to a stiff and homogeneous gel embracing all the constituent elements of the reacting solutions. This gel has such great rigidity of structure, and is relatively so low in water content—comparing favorably in this respect with a gelatinous precipitate which has been vigorously filter-pressed—that it may be dried directly. It then breaks down into small lumps which, when placed in water, granulate into particles of a size frequently suitable for water softening without crushing or grading.

The fact that a granule of this type of zeolite is a fragment of a strong and homogeneous structure and not a compacted agglomeration of small precipitated particles, is undoubtedly responsible for some of its distinctive properties and probably accounts for its classification as the gel type.

One of the advantages of zeolites of the gel type lies in the fact that there is in them a large latent capacity which may be called into service by the expenditure of more than the most economical amount of salt. In consequence a softening element using a gel-type zeolite is capable of providing for a considerably increased demand for soft waters simply by increasing the amount of salt employed. The author discusses next certain factors in the operation of zeolite-type water softeners, of which the following only will be mentioned.

The amount of water required for washing the zeolite bed free from the brine used in regeneration is often quite an important item in operating expense. It appears to depend entirely on the area of the bed, the depth of bed being immaterial within the usual limits; the smaller area possible with the gel-type zeolite accordingly reduces the water consumption for this purpose.

In downflow softening—that in which the water being softened passes through the zeolite bed downwardly—an extremely important engineering factor is the loss of pressure through the zeolite bed. With some types of zeolites this loss of pressure is frequently so serious that the additional pumping necessary in consequence is an appreciable item. With a gel-type zeolite the loss of head is relatively quite small. The reason for the difference is easily found. The particles of green sand are quite small—practically all are finer than 20 mesh, and particles down to 60 mesh are commonly employed. The granules are rounded and rather regular. The combination of these two characteristics accounts for the decided tendency of a green-sand bed to pack and to "channel"—that is, to break through at points of least resistance in the bed. With the gel zeolite the particles are relatively large—the material used in the tests shown is 8 to 50 mesh—and are irregular and sharp-edged. The packing tendency is thus minimized.

As Siedler, one of Gans' co-workers, pointed out, softening with zeolites may be either upward or downward. This is particularly interesting in view of some court decisions in this country which indicate that downward softening is the only feasible method. Seidler was right. As a matter of fact, for fairly soft waters upward softening is the only practical method, particularly in using

high-capacity, rapid-rate zeolites of the gel type. In the case of extremely hard waters, this has not yet been shown to be true.

Several months ago several large softening units were installed in Chicago, all at about the same time, to operate with the gel-type zeolite referred to on Lake Michigan water, which has a hardness of 8 grains per gallon (135 p.p.m.). Using downflow softening, for which these plants were originally designed, it was found that a comparatively low total base-exchange capacity could be obtained with economical salt consumption. The direction of flow was then reversed so that the water passed upwardly through the zeolite bed, whereupon the softening capacity was increased by one-third, with the same salt consumption per unit of hardness as before. These improved results have been maintained consistently since the change was made. Another advantage of upflow softening was that the loss of pressure through the zeolite bed was practically zero. Rates as high as 10 to 12 gal. per sq. ft. per min. could be satisfactorily employed. (A. S. Behrman, International Filter Co., Chicago, Ill., in *Industrial and Engineering Chemistry*, vol. 19, no. 4, Apr., 1927, pp. 445–447, 3 figs., *d*)

Erosion Caused by Jets

THE present article deals with tests on erosion carried out with varying steam conditions with the view to determine the factors affecting erosion of turbine blades.

The steam conditions are given in Table 9. The materials investigated were monel metal, sheet nickel, and others, including

TABLE 1 STEAM CONDITIONS DURING EROSION TESTS

Test No.	Steam temp. and pressure in front of nozzle: deg. cent.	Steam temp. and pressure in front of nozzle: kg. per sq. cm.	Steam pressure after nozzle, kg. per sq. cm.	Adiabatic drop, cal.	Dryness fraction	Velocity: Theoretical m. per sec.	Velocity: Actual m. per sec.
1	113	1.19	0.294	63	1.00	727	699
2	220	1.58	0.274	65	0.962	738	709
3	160	1.51	0.271	69	0.962	717	731
4	139	3.54	0.190	110	0.876	958	921
5	177	9.68	0.161	151	0.829	1123	1080
6	180	10.00	0.110	163	0.817	1170	1120

several kinds of rustless steel and a 5 per cent nickel steel. Absolutely no erosion occurred during tests Nos. 1 to 4, and only after test No. 4 was the surface of monel metal and nickel steel roughened due to erosion. A further increase in the steam velocity then produced an appreciable degree of erosion.

It is very remarkable that the slight increase in the steam velocity from 1080 to 1120 m. per sec., with an increase of 7.0 per cent in the wetness, results in erosion taking place at a very much greater rate. It is probable that a further increase in the velocity of the steam would be accompanied by still more rapid erosion. The available apparatus, however, did not permit of any further extension in this direction. The great importance of the velocity is also shown by tests Nos. 1 to 4 in which erosion worthy of mention did not occur, although the remaining conditions were similar to those in tests Nos. 5 and 6, the velocity of the steam only having been reduced by about 20 to 40 per cent. This reduction was sufficient to prevent all erosion during the seventy hours' testing period. It is noteworthy that even the reduced speeds were always very great; in any case they exceed the probable values for the water drops striking the blades in a steam turbine.

Tests with steam jets do not allow the phenomena to be closely analyzed. The conditions prevailing in the steam jet are as yet only imperfectly known; the water content of the steam cannot be determined in a reliable manner. But little can be accurately stated about the distribution of the water in the steam, i.e., the size of the drops. The speed to which the water is accelerated by the expansion of the steam is not known—it can only be safely said that the water speed is rather less than the speed of the steam. As the size and velocity of the drops exert an exceedingly great influence upon the erosion, the nozzle research offers no explanation of the importance of the various individual factors.

These defects are obviated by a recent method of testing. The metals to be tested are made into cylindrical or prismatic rods and screwed into the rim of a wheel which is driven at various speeds by means of an electric motor. At the maximum speed the test pieces run at a velocity of 225 m. per sec. (721 ft.). A fine jet of water, parallel to the shaft of the impeller, crosses the path of the test pieces and strikes each piece once per revolution (Fig. 9). The test pieces are spaced around the rim so that each is struck by an unbroken jet. The tests differ from the conditions prevailing in the last stages of steam-turbine blades in that the drops of water striking the metal are much larger; in fact, the erosion produced during these tests is much more rapid than that met in practice. A few hours in the testing plant suffice to cause erosion to a depth of a millimeter, even with the hardest materials. It is thus possible, in a short period, to determine the resistance of different metals

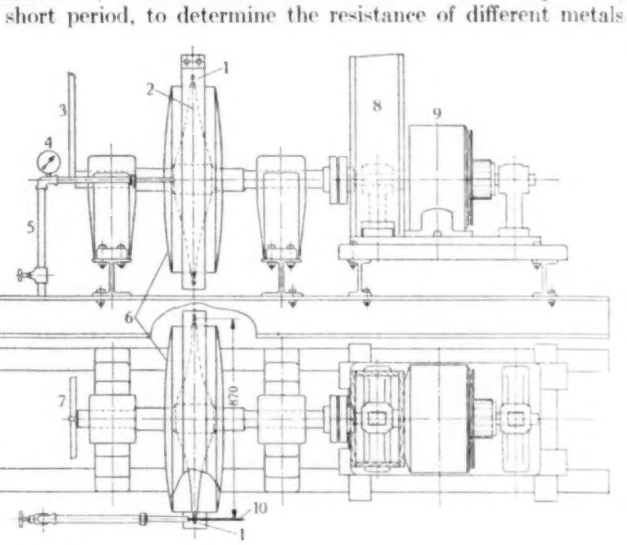

FIG. 9 ARRANGEMENT OF PLANT FOR TESTS USING WATER JET

(1, Specimen; 2, disk; 3, belt to tachometer; 4, gage; 5, water main; 6, casing; 7, tachometer belt; 8, ventilation shaft for motor; 9, motor; 10, water jet.)

under the action of erosion. It remains to be provided that the results obtained under these severe conditions may be directly applied to steam turbines in which the conditions are much less strenuous. This assumption will be made for the time being. The results obtained in practice and those from testing agree so well with each other that the assumption has been completely justified at least qualitatively, i.e., a list comprising metals in order of their resistance to erosion holds for practical and testing purposes, no matter whether the conditions to which the specimens are subjected are severe or not.

The purely qualitative tests have shown that erosion increases rapidly with the velocity. Metals which resisted erosion at a speed of 130 m. per sec. (426 ft.) showed large holes after having been tested at 200 and 225 m. per sec. (656 and 738 ft.). Further tests would indicate that reduction in weight caused by the impact between the jet and the specimen is a function of the total number of impacts.

Another series of tests shows the relation existing between the resistance to erosion of rustless steel and the heat treatment, hardened steel showing decreased erosion losses. The article gives data showing the behavior of various metals with respect to erosion. Among other things, it has been found that cast iron offers a smaller resistance to erosion than steel. The size of the drops of water appears to have a very material effect. By electrolytically chrome plating 5 per cent nickel steel, the resistance to erosion is considerably increased. On the whole, it has been found that at first the specific erosion increases with the number of impacts, but finally decreases.

The behavior of the material during the tests may be explained as follows: As long as the surface of the specimens is smooth, it offers an unfavorable surface for the impinging water to attack, hence the water flows off to either side. Erosion, therefore, does not take place for some time. As soon as any roughness is formed, however, the erosion proceeds rapidly as the jet impinges with great force in the unevenness. If, finally, the unevenness has attained a considerable depth, a layer of water adheres to the now completed roughened surface. This water absorbs part of the impact of succeeding drops of water so that the force of the jet is not so effective as formerly. The specific erosion consequently decreases after a certain depth has been attained. (*Brown Boveri Review*, vol. 14, no. 4, Apr., 1927, pp. 95–104, 14 figs., *eA*)

RAILROAD ENGINEERING

The Kitson-Still Locomotive

DESCRIPTION of a locomotive said to be now under construction, powered with a Still engine. For a general description of the Still engine see MECHANICAL ENGINEERING, vol. 44, no. 5, May, 1922, p. 316.

The engine is double acting with internal combustion at one end of the cylinder and steam at the other end, through which the piston rod works. The water in the jacket is in connection with the boiler, and the excess heat from products of combustion assists in the production of steam in the boiler. The boiler is heated by oil burners and the steam generated is used for starting the engine.

The locomotive is of the 2-6-2 type, calculated to exert a tractive effort of 24,500 lb. from starting to a speed of 6 m.p.h., dropping to 7000 lb. at 45 m.p.h. The engine is an 8-cylinder 4-stroke-cycle horizontal type. Impulses are transmitted first to a crankshaft mounted in rigid bearings and then by rotating gearing protected by a spring and damping device to the wheels through coupling rods. The internal-combustion engine is directly reversible. The details of construction are given in the original article. It is expected that the first cost will be on a parity with a steam unit of the same power. The consumption of water in this engine is one-tenth that of a steam locomotive of similar power in equivalent service, which might prove to be of particular value in countries where the matter of supply of water is serious.

In the discussion which followed, Capt. H. P. Beames anticipated defects in regard to the transmission due to the rise and fall of the jackshaft in consequence of the interposition of the spring system. He also said that experience with oil fuel had so far shown that it was not at present a commercial proposition in England. (Paper by Lt.-Col. C. Kitson Clark before the Institution of Mechanical Engineers, London, April 8, 1927, abstracted through *The Railway Gazette*, vol. 46, no. 15, Apr. 15, 1927, pp. 501–503, *d*)

SPECIAL MACHINERY

A Four-High Continuous Strip Mill

IN CONNECTION with the article on Recent Developments in the Rolling of Sheet Steel in MECHANICAL ENGINEERING, vol. 49, no. 5, May, 1927, pp. 451–454, the following two items may be of interest.

The Laclede Steel Co. now has under construction at its Alton, Ill., plant one of the latest-type units. The mill will apply the principle of 4-high rolls to the manufacture of thin strip steel. The 4-high principle of rolling has been tried with success in the manufacture of wide plates, sheets, and strip sheets. In the case of the Alton, Ill., mill, the procedure, starting with the heating of a billet, slab, or bloom in a continuous furnace, will be to lead it through an edging pass; cut to the length required with a flying shear; break down in four stands of roughing rolls; pass through an intermediate stand; finish through five 4-high roller-bearing sets of finishing rolls; cut to length with another flying shear; and go through the runout on to a specially designed hot bed.

The three edging stands of rolls will be driven independently by direct-current motors; the roughing, intermediate, and the No. 6 stands of rolls will be driven by one motor; while the 4-high finishing stands will be driven by independent direct-current motors, permitting individual speed adjustments. The dominant feature in the design of the runout and hot beds is to effect alignment and flatness in the finished strip. (P. 1124 of *Iron Trade Review*, Apr. 28, 1927.)

A 4-high mill with the back-up rolls mounted in roller bearings recently was purchased by the Newton Steel Co. for its works at Newton Falls, Ohio. The mills will be installed in the cold-rolling department. Three sheetmakers in other districts also are planning to install mills of this type. (*Iron Trade Review*, vol. 80, no. 17, Apr. 28, 1927, pp. 1124 and 1099, *g*)

SPECIAL PROCESSES

De Vecchis Process for Making Beet Sugar

THIS process has been considered by the British Ministry of Agriculture and Fisheries and an experimental unit has been installed and operated by the Institute of Agricultural Engineering of the University of Oxford. One of the essential elements in this process is the drying of beets and as at the University of Oxford a considerable amount of work was done previously on drying of agricultural products, some of the results obtained have been applied to the new process of beet-sugar agriculture.

In the De Vecchis system of drying as now used the cossettes (briquets) of dried material are produced in less than one hour, about 75 per cent of the total weight being removed, reducing the moisture content to about 5 per cent. It was found essential to remove the moisture quickly so that the material should not be heated in a massed state for any appreciable length of time. Caramelization occurs at definite temperatures varying with the amount of moisture present. Although it may not cause a large direct loss of sugar, caramel has pronounced coloring properties which result in difficulties in the after-treatment of the juice. Improper drying may also produce inversion, which is a hydrolyzing action that transforms sucrose into what is known as invert sugar, a non-crystallizable substance, so that when inversion takes place there is a direct loss in the amount of sugar produced. In the Oxford process it was found that so long as the temperature to which dry or partially dried cossettes were exposed did not exceed 110 deg. cent. (230 deg. fahr.) no formation of invert sugar or caramel occurred. The apparatus developed (which cannot be described here because of lack of space) works on the principle of mass drying, i.e., removal of water by drying the material in a heavy layer or mass which has consolidated owing to its initial inherent weight. The character of the air conveyed to the mass is carefully determined in order to procure, first, the natural exudation of water due to weight; second, the greatest quantity of water taken away by the carrying property of the heated air; and third, the natural heat reaction which takes place within the vegetable material when heated.

The essential element in the new process therefore lies in the desiccation system, which is said also to make the process of purification simpler. As practiced at Oxford, the latter process was modified so that the juice is first treated with lime and the excess of that substance is subsequently neutralized with calcium superphosphate. At first some difficulty was experienced with the filtration of the thick juice, but it was found that short treatment in a centrifuge removed such a large percentage of the suspended bodies that filtration was rendered easy. Estimated costs of installation and operation of a factory working on this process are given in the original article. (*The Engineer*, vol. 143, no. 3719, Apr. 22, 1927, pp. 431–434, 7 figs., *d*)

TESTING AND MEASUREMENTS (See Welding: Non-Destruction Tests for Welds)

THERMODYNAMICS

Some Problems in the Conduction of Heat

MANY of the every-day problems in heat conduction involve the transference of heat from one medium to another. Problems of this type are as a rule difficult to solve by the methods usually given in textbooks. In the present paper attention is called to a method specially suited to such problems, and the fundamental solutions required in applying the method are given. The main idea underlying the method employed is to treat all heat-conduction problems as problems in wave motion. Instead of building up the solution required from instantaneous point sources or doublet sources the author builds them up from wave trains, and in this way the methods and principles employed in the solution of wave-motion problems become available for heat conduction. He begins by showing how the well-known expressions for the instantaneous-point-source and doublet-source solutions can be obtained by means of wave trains.

The article is not suitable for abstracting, but the following is given as illustrating the fundamental principles used. The general equation of conduction of heat in a uniform medium may be written in the form

$$\frac{\partial v}{\partial t} = \kappa \nabla^2 (v) \qquad [1]$$

where v represents the temperature and κ the diffusivity. There are two solutions of the situation representing wave trains traveling in the direction of x positive, namely,

$$e^{-x\sqrt{\frac{k}{2\kappa}}}\cos\left(kt - x\sqrt{\frac{k}{2\kappa}}\right)$$

and

$$e^{-x\sqrt{\frac{k}{2\kappa}}}\sin\left(kt - x\sqrt{\frac{k}{2\kappa}}\right) \quad \ldots\ldots\ldots\ldots\ldots [2]$$

By changing x into $-x$ are obtained the corresponding solutions representing wave trains traveling from the origin in the direction of x negative, and from this may be obtained more general solutions of the same type by summations or integrations after the manner of Fourier. The author gives examples of this development. (Dr. Geo. Green, University of Glasgow, in *London, Edinburgh and Dublin Philosophical Magazine and Journal of Science*, vol. 3, no. 16, supplementary number, April, 1927, pp. 784, 800, *pm*)

TRANSPORTATION

An Aluminum Trolley Car in Cleveland

Description of the car built by the Cleveland Railway, chiefly of duralumin. The standard Cleveland drawings for steel cars were followed, the duralumin being substituted for steel with practically no changes in section sizes except that the size was increased for the body side sheets, the truck side frames, and the body bolsters. The side sill angles and some other few minor parts are of steel. The car weighs 30,300 lb. as compared with 43,200 lb. for the company's standard car of similar dimensions. Since this car was built entirely on an experimental basis, no comparative cost figures are available. In general, the aluminum alloys weigh about one-third as much as steel, and the cost per pound in structural form is roughly six times that of steel. For a given structure, therefore, the cost of duralumin would be about double the material cost in steel, but against this should be credited the scrap value of the aluminum alloy. No figures are given as to comparative power consumption by this car.

Except for the windows, doors, side-sill angles, wheels, axles, springs, gears, journal boxes, air tanks, resistor grids, trolley base, motor magnet frames, and the main body of the air compressor, the car structure and equipment are built almost entirely of duralumin. The material was furnished by the Aluminum Company of America. Rolled plates and extruded structural sections are made of what is known as 17-S alloy tempered to an ultimate strength of 55,000 to 60,000 lb. per sq. in., yield point 30,000 to 40,000, elongation 18 per cent to 20 per cent, and modulus of elasticity 10,000,000. The material used in forgings is called 15-S alloy and has practically the same physical characteristics as 17-S alloy except that the elongation is 16 per cent to 18 per cent instead of 18 per cent to 20 per cent. (*Electric Railway Journal*, vol. 69, no. 15, Apr. 9, 1927, pp. 655–658, *d*)

An Electric Trolley Bus

The first pneumatic-tired Guy six-wheeled double-deck electric trolley bus has just been put in service at Wolverhampton. It has a covered top, a rear stairway, and seating accommodation for sixty passengers, constituting a serious rival to the tram car. The electric motor develops 60 hp. at 900 r.p.m. and is a new design incorporating many improvements in this class of motor. The six-wheeled chassis follows the standard Guy design, having a dropped frame which gives a low loading line so that only one step is required. The Wolverhampton Corporation have placed a repeat order for ten more of this type. (*British Commercial News*, Bulletin A, No. 25, April, 1927, p. 14, 1 fig., *d*)

WELDING

Non-Destruction Tests for Welds

The following three tests have been considered.

1 *The Electric-Resistance Method.* The electric resistance across a weld can be accurately measured even when the weld is part of a structure, such as a steel ship. The difficulty lies, however, in finding a relation between the resistance and the strength of the weld, and this has not yet been done.

2 *Magnetic Analysis.* In this case variation of magnetic flux under a constant magnetizing force is considered. The process is inexpensive, rapid, and can be used almost anywhere, but does not infallibly indicate defects that affect the strength of the joint. It is being used to locate defects in steam-turbine bucket-wheel forgings. It is difficult, however, to see how it can be used on welds where the thickness and contour of the metal vary greatly, such, for example, as on the fillet weld where an I-beam joins a column. (see also at the end of this abstract reference to work along these lines by the U. S. Bureau of Standards).

3 *X-Ray Examination.* The work that has been done on this method seems to show that many defects in welds can be detected, but the defects must be cavities in the metal either filled with gas or non-metallic inclusions, or must be cracks or other discontinuities that are approximately parallel to the direction of the rays. The indications, fortunately, are not affected by the stress in the material or by variations in the heat treatment. At present it is evident that more work should be done to perfect this method, as it is not affected by conditions that are unavoidable in welds. One may therefore expect within a few years that X-ray examinations will be made commercially on important welds if the thickness does not exceed two or three inches. The cost, however, is unduly high at the present time. (Dr. H. W. Whittemore in a paper before the International Acetylene Association, abstracted through *Mechanical World*, vol. 81, no. 2102, Apr. 15, 1927, pp. 265–266, *p*)

In this connection attention is called to the U. S. Bureau of Standards Scientific Paper No. 546. (Compare *The Iron Age*, vol. 119, no. 17, Apr. 28, 1927.)

In connection with research of the relationship which exists between the magnetic and mechanical properties of iron and steel, the United States Bureau of Standards has discovered that the resistance which steel offers to magnetization does not truly represent the magnetic properties of the material, and therefore cannot be expected to serve as a basis for correlating them with the mechanical properties of the metal. The research has been conducted by the bureau in an effort to discover a definite connection between the two sets of properties in steel.

If such a relationship as the one sought could be established, according to the bureau, non-destructive methods of great value could be developed for the testing of steel and steel products. One of the most interesting subjects of scientific research at the present time is the relationship between these properties, and, in the effort to discover a definite connection between them, various ways of expressing the magnetic characteristics of a material have been developed.

One of these ways is by means of the so-called reluctivity relationship, which represents the magnetic reluctivity (resistance to magnetization) as depending directly in a simple way upon the intensity of the magnetizing force. The equation representing this relationship contains two numerical constants. If these constants are truly characteristic of the material and if there is a definite relationship between magnetic and mechanical properties, they should be useful for purposes of correlation and ultimately form the basis of a practical method of test.

The experimental and theoretical study recently carried out at the Bureau of Standards has led to the conclusion that this reluctivity relationship does not truly represent the magnetic properties of the material, and therefore cannot be expected to serve as a basis for correlating them with the mechanical properties. Therefore some other basis of correlation must be sought. Although the work of the bureau has not succeeded in developing a method by which mechanical properties can be determined from a knowledge of the magnetic properties, it has been useful in showing what not to expect.

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Arteiles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

The Conference Table

THIS Department is intended to afford individual members of the Society an opportunity to exchange experience and information with other members. It is to be understood, however, that questions which should properly be referred to a consulting engineer will not be handled in this department.

Inquiries will be welcomed at Society headquarters, where they will be referred to representatives of the various Professional Divisions of the Society for consideration. Replies are solicited from all members having experience with the questions indicated. Replies should be as brief as possible. Among those who have consented to assist in this work are:

ARCHIBALD BLACK,
Aeronautic Division
H. W. BROOKS,
Fuels Division
R. L. DAUGHERTY
Hydraulic Division
JAMES A. HALL,
Machine-Shop Practice Division
CHARLES W. BEESE,
Management Division
G. E. HAGEMANN,
Materials Handling Division
J. L. WALSH,
National Defense Division
L. H. MORRISON,
Oil and Gas Power Division
W. R. ECKERT,
Petroleum Division
F. M. GIBSON and W. M. KEENAN,
Power Division
WINFIELD S. HUSON,
Printing Machinery Division
MARION B. RICHARDSON,
Railroad Division
JAMES W. COX, JR.,
Textile Division
WM. BRAID WHITE,
Wood Industries Division

Aeronautics

Forced-Landing Causes[1]

A-17 What are the causes of forced landings in airplane operation?

Forced landings in the operation of airplanes occur for various reasons. Chief among these is weather. A better system of aviation weather reports would, in the writer's opinion, eliminate a considerable proportion of the forced landings from this cause. Mechanical failure is often a contributing factor. In the latter classification the major portion of the trouble is with power-plant installation or accessories, rather than any trouble inherent in the motor. There is more trouble with the plumbing system, ignition, carburetion and fuel and oil supplies than with any actual breakage of parts of the engine. The forced landings from mechanical failure are becoming fewer and fewer all the time. This is partly due to better equipment, and partly to better installation and maintenance. (Richard H. Depew, Jr., Vice-President, Fairchild Flying Corporation, New York, N. Y.)

Fuels

F-14 In determining the loss due to combustibles in flue gases, what are (*a*) suitable methods of taking average samples of gases for determining the content of dust therein, (*b*) filters required for this purpose, and (*c*) methods for ascertaining both the quality and quantity of hard particles gathered?

No method has been standardized, and that used must be adjusted to conditions and the purpose for which the data are required. Quantitative sampling to determine the percentage of solids carried in the flue gases is difficult to accomplish accurately, because of the uncertainty of the uniformity of the dust distribution over a cross-section of the flue. It is therefore necessary to explore the area by selecting a sufficient number of sampling points if fairly accurate data are required.

1 The piping through which the gas is drawn from the furnace should be smooth and free from joints which might catch and hold the dust. It should be small enough to give a high velocity in the pipe at the minimum rate at which the gases are drawn off. The end in the flue through which the gases enter should face the gas stream, and the volume drawn in should be adjusted so that the velocity of entrance approximates that of the gas stream at the sampling point. The steam velocity can be determined by a pitot tube. The filtering arrangement should be located so that the piping is as short as possible. The quantity of gas drawn for a single sample and the suction necessary will depend on the type of dust and on its clogging effect on the filter. It is more convenient to use a positive-exhaust pump, since this type gives higher suction ranges. The quantity of gas drawn is best measured by an orifice placed after the filter, since it can be also used to keep the flow constant. The corrosive nature of the gases is hard on meters. It is important to keep the gases above their dewpoint temperature until they have passed the filter, otherwise the condensed water holds the dust in the piping and clogs the filter.

2 A convenient filtering arrangement can be made so as to include the centrifugal principle to collect the heavier particles and a paper thimble to catch the fine dust. A suitable design cannot be shown without a drawing, but it will consist of a glass vessel, such as a clear cylindrical electric-bulb glass shade, fitted with an air-tight cover, into which will be fastened an inlet pipe arranged so as to swirl the gases, and an outlet pipe carrying the paper-thimble support.

3 The quantity of dust collected will be determined by weight, and the quality by chemical analysis or microscopical examination. If only the percentage of combustible is required, it can be determined by proximate analysis as for coal, and assumed to be carbon.

In the foregoing the gases have been assumed to be free from tar and soot. If these are present the difficulties are increased, due to their tendency to deposit in the piping, and because they clog the filter and thus limit the quantity of gas which can be drawn through without renewing it.

The method outlined is probably the simplest, but the results obtained must be considered indicative rather than an accurate measure of average conditions, unless a large number of samples are obtained. (P. Nicholls, Supervising Fuel Engineer, U. S. Bureau of Mines Experiment Station, Pittsburgh, Pa.)

Machine-Shop Practice

Machining Copper[1]

MS-11 What lubricants and coolants should be used in machining pure copper?

In the practice of the writer's company, signal oil has been used for planing long, broad surfaces where the highest grade of smooth surface obtainable is required. On other work, lard oil cut with kerosene is used. This also applies to milling and turning. For drilling, water or kerosene is used, if it is not deemed advisable to use the lard or signal oil. (Sidney R. Jones, Designer and Draftsman, Anaconda Copper Mining Co., Great Falls Reduction Dept., Great Falls, Montana.)

Measuring Cutting-Tool Temperatures

MS-12 Have any of the members knowledge of methods of determining the temperature rise in machine cutting tools other than the usual method of observing temperature of coolant or change in color of tool? A brief description of such a method will be appreciated.

A number of investigators have attempted using the tool and the work as a thermoelectric couple, and measuring the temperature at the point of the tool on the proper electrical measuring instruments. This has been done in various ways, and is described in the following articles:

1 Thermoelectric Measurement of Cutting-Tool Temperatures, Henry Shore, *Journal*, Washington Academy of Sciences, March 4, 1925, p. 85.

2 A New Method of Measuring Temperature Generated by

[1] This subject has been discussed in a previous issue.

Metal Cutting Tools, Edward G. Herbert. (Thermoelectric, stellite tool cutting mild steel.) Description, *The Pendulum*, June, 1925. Some data, *The Pendulum*, September, 1925.

3 The Measurement of Cutting Temperatures, E. G. Herbert. Paper, Institution of Mechanical Engineers, February 19, 1926.

4 Supplementary paper on the Measurement of Cutting Temperatures, E. G. Herbert. *American Machinist*, March 4, 1926, p. 363.

Another article on this general subject is Die Messung der Schneidentemperature beim Abdrehen von Flusseisen (Measuring the Cutting Temperatures in the Cutting of Low-Carbon Steel), Gottwein, *Maschinenbau*, November 19, 1925, pp. 1129–1135. The writer is not sure what methods are followed in this article, however. (James A. Hall, Professor of Mechanical Engineering, Brown University, Providence, R. I.)

Power

P-3 How successful in operation in industry has been the electric boiler?

(*a*) Electric boilers have proved successful in operation in all cases where a careful analysis has been made beforehand of their adaptability. This analysis must give consideration of such factors as cost of power, location of equipment, type of control, type of power contract, etc. (E. H. Horstkotte, Industrial Engineering Department, General Electric Company, Schenectady, N. Y.)

(*b*) In recent years there has been very little published concerning the use of the electric boiler in industry. However, the number of manufacturers of this type of boiler is increasing. In the 1926 E.M.F. Yearbook 16 manufacturers of electric boilers are listed.

P. H. Falter, vice-president of the Electric Furnace Construction Company, Philadelphia, Pa., describes several electric-boiler installations.

1 The largest installation consists of two 25,000-kw. boilers, 6600 volts, 3-phase, 60 cycle. Each boiler delivers 77,500 lb. of steam per hour at 135 lb. pressure. This is located at Grand Mere, Quebec, the Laurentide Company. One man per shift handles the boiler operation, while with coal-fired boilers 37 men are required to get the same steam generation from 14 boilers. Another item is the saving of approximately 50,000 tons of coal per year.

2 At the Brown Company, Berlin, N. H., one man handles three 18,000-kw. boilers generating 55 tons of steam per hour at 125 lb. pressure. Current is used at 22,000 volts, 3-phase, 60-cycle.

3 The Niagara Falls Power Company generates 54,000 lb. steam per hour with one 18,000-kw. boiler.

4 The Ford Motor Company at Green Island, N. Y., uses two electric water heaters each rated at 3000 kw., at 4600 volts, 3-phase. The following installations are listed for 1923:

	No.	Rating of each, kw.		No.	Rating of each, kw.
Clinton, Mass...	1	40	Green Island.....	1	3200
Buffalo, N. Y....	1	100	Quebec..........	1	5000
Peterborough, N. H..........	1	130	Quebec..........	1	5000
Watertown, N. Y.	2	1000 and 350	Port Angeles, Wash..........	1	5000
Millers Falls, N.Y.	1	350	Niagara Falls, N. Y..........	1	6000
Quebec..........	1	350	Kenogarni, Can...	1	7000
Niagara Falls, N. Y..........	2	18000 and 1000	Pittsburgh, Pa....	1	7000
Quebec..........	2	1300	Quebec..........	1	12000
Quebec..........	2	1500	Rumford, Maine..	1	16000
Rochester, N. Y..	1	3000	Berlin, N. H.....	1	18000
Ontario..........	1	3000	Quebec..........	1	20000
			Quebec..........	2	25000

The main use of electric boilers is found in paper and pulp plants and electrochemical plants near hydroelectric developments. In the St. Lawrence region hydroelectric power is comparatively cheap, while coal is high. Therefore, it is more economical to generate steam for heating and process work by electricity, rather than in coal-fired boilers. In large industrial plants using a large amount of electricity it is often economical to use electric boilers for steam generation in locations far distant from the main power plant. On electrified sections of trunk railroads a small electric boiler is often used on the locomotive to furnish steam for heating the cars.

In conclusion, the electric boiler has proved successful in industry where conditions of location have made it more economical than the coal-fired boiler. In certain cases where large blocks of electric power are purchased, electric boilers can be used to advantage. They are new, and whether any troubles, such as the formation of hydrogen gas in the boiler, will develop is not known. (Dudley P. Craig, Instructor, Mechanical Engineering, Purdue University, Lafayette, Ind.)

Miscellaneous

Gas Inlets to Shell-and-Tube-Type Ammonia Condensers[1]

M-3 In vertical shell-and-tube-type ammonia condensers, what have operation results indicated to be the most desirable point of admission of the gas into the shell?

Admission should be near the top of the shell, and, in the opinion of the writer, the inlet nozzle should be tangential to the shell, a suitable baffle being provided to protect the tubes nearest the nozzle from direct impingement of entering gas. It is important that the non-condensable gases accumulate at the bottom of the condenser, from which point they can readily be purged; hence it is important that the gas be introduced near the top of the shell. (S. C. Bloom, Consulting Engineer, Chicago, Ill.)

Liquid-Ammonia Pump for Cold-Storage Plant[1]

M-4 In a certain electrically driven cold-storage plant having the ammonia condensers located near the ground floor it is necessary in the winter to reduce the condenser surface to create sufficient pressure to deliver the liquid ammonia to the evaporating coils in the upper-floor rooms. The difference in head pressure between summer and winter operation is about 50 lb. per sq. in., corresponding to a head pressure of approximately 200 ft. of liquid ammonia. Would it be practicable from the standpoint of maintenance to install a liquid-ammonia pump to supply the coils and allow the compressors to operate on a lower head pressure with full condenser cooling surface? The pump would be required to handle a maximum of 50 gal. per min. against a head of 125 to 150 lb. per sq. in., including friction, with a suction head of about 75 lb. Could such a pump be made automatic, or pressure-controlled, so that it would need to run only when the pressure conditions were such that insufficient refrigeration was being obtained on the upper floors?

(*a*) It is manifestly inefficient to maintain an unduly high head pressure in order to force liquid ammonia to the evaporating coils. The writer is of the opinion that it would be practicable to install a liquid-ammonia pump as a booster to supply the coils, and operate the ammonia compressor at the lowest possible condenser pressure.

The specific case which has been cited is evidently a large plant, as 50 gal. per min. of liquid ammonia is over 300 lb. per min., equivalent to approximately 750 tons of refrigeration. The writer has not used liquid-ammonia pumps with automatic controls, but sees no reason why such control could not be used. If, the pump is made no larger than necessary, it might be arranged to run fairly continuously during the season when the pressure conditions require its use. As the writer sees it, the use of the pump would be restricted to certain seasons of the year, unless the plant uses well water, in which case the head pressure would be more nearly constant, unless the condensing surface is insufficient. A recently patented electromagnetic pump might be worth consideration for this service. Also, it might be well to communicate with C. N. Smith, Chief Engineer, Sheriff Street Market and Cold Storage Co., Cleveland, Ohio, who has employed a booster pump somewhat as described. (George A. Horne, Manager, Technical Department, Merchants Refrigerating Co., New York, N. Y.)

(*b*) About $3^1/_2$ hp. will be required to do this work. It will pay to use a separate pump to force liquid to top of building and keep condenser pressure down. The pump could be equipped with a low-pressure cut-in and high-pressure cut-out to work automatically. (L. S. Morse, Chief Engineer, York Manufacturing Company, York, Pa.)

(*c*) The writer has had no direct experience with liquid-ammonia pumps, but no doubt a pump of the centrifugal type could be ob-

[1] This subject has been discussed in a previous issue.

tained for this service. The feature that would require close inspection would be the shaft packing. A large receiver placed in the liquid line at a height to overcome the difference in head between the condensing and evaporating pressures with a pressure-regulated electric switch to control the operation of the electrically driven pump would be the only auxiliary equipment required. An outlet should be provided on the top of the receiver connecting into the suction side of the system with a manually operated valve to permit purging out accumulated gases from this point. A bypass line should be placed around the pump so that with a pressure on the condensing side sufficient to lift the liquid to the required height the pump would not operate. This bypass line should be equipped with a check valve to prevent backing up of the ammonia into the regular receiver when the pump is operating.

The above presupposes that a pump is necessary; however, assuming a temperature of the liquid ammonia of 55 deg. fahr. and a gage pressure of 83.4 lb., one pound pressure would elevate the liquid 44.6 lb., or 3.716 ft. A head pressure of 125 lb. static would be 125 × 3.716 or 465 ft.

If the condensing pressure is 83.4 lb. gage and the evaporating pressure 30 lb. gage—the usual extreme pressures in a refrigerating plant—the difference in pressure is 53.4 lb., which is sufficient to raise the liquid ammonia 53.4 × 3.716 = 198 ft., not considering friction of the liquid in the pipe. (J. O. Schultz, Chief Engineer, The Triumph Ice Machine Co., Cincinnati, Ohio.)

Questions to Which Answers Are Solicited

Power

P-4 To what extent is hydrochloric acid being used for cleaning condenser tubes, and what have been the experiences of members with this method of cleaning?

Fuels

F-15 In a pulverized-fuel-fired boiler furnace the finer the pulverization, other things being equal, the more complete is the combustion in a given furnace. Conversely, however, finer pulverization requires more power per ton of coal pulverized. What have operators and manufacturers found to be the most economical fineness of pulverization (*a*) in unit-fired installations and (*b*) in central-plant installations? Describe also conditions under which these results have been obtained as to maximum heat released per cubic foot of furnace volume, fusion temperature of ash used, etc.

Correspondence

Contributions to the Correspondence Department of Mechanical Engineering are solicited. Contributions particularly welcomed are discussions of papers published in this journal, brief articles of current interest to mechanical engineers, or comments from members of The American Society of Mechanical Engineers on activities or policies of the Society in Research and Standardization.

The Term "Heat Cycle" a Misnomer

To the Editor:

On page 643 of the Transactions of the A.S.M.E. for 1923 the term "heat cycle" occurs, and on page 1420 of the December, 1926, issue of Mechanical Engineering the same term again appears. So far as I have observed, these are the only two cases in which this term has been used in any of the publications of the Society. If the editors, or whoever is responsible, can justify the use of such a term, I should appreciate having the explanation.

The term "heat cycle" seems to me to be a misnomer and I believe its use should be discouraged in the Society, which aims to produce good technical literature. Engineers are constantly dealing with *steam* cycles and with the cycles of the various pieces of apparatus through which the working substance passes, but when do they ever work with "heat cycles?"

The entire justification in building any steam power plant lies in the expectation that such a plant will be successful in *transforming some of the heat* evolved from combustion of the fuel into mechanical energy, and thus the thought of having a "heat cycle," or in other words in making the heat complete a cycle is contrary to the basic purpose for which the plant is built. In many modern plants it is perfectly true that a small portion of the steam passing the throttle may be bled from the prime mover *after delivering some of its energy to the turbine*, but does this justify the expression, "The main turbines will be bled in order to complete the heat cycle?" (See p. 1420, Mechanical Engineering, December, 1926.)

The passage of the working substance through its cycle involves many more processes than its mere flow through the prime mover, but both cases are very important in modern power-plant engineering and each can be completely and easily analyzed; however, different energy terms are involved. On the other hand, who can furnish an analysis of a "power-station *heat* cycle," or who can even show that there is such a thing in any power station?

This letter is written solely for the purpose of trying to keep our technical literature up to the high standard that all of us hope for. If it succeeds in reducing somewhat the enormous overload on the already much abused term "heat," it will have served its purpose.

Frank O. Ellenwood.[1]

Ithaca, N. Y.

[Professor Ellenwood's letter was submitted to C. Harold Berry, Associate Editor of *Power*, for comment, and the following supplementary discussion was contributed by him.—Editor.]

To the Editor:

I am heartily in accord with the remarks of Professor Ellenwood. A cycle is a process in which a material body standing by itself, or typical of a continuous stream of substance, passes through a series of states, periodically reproduced. During its execution of the cycle the body at times absorbs heat, again develops heat, and likewise alternately absorbs and develops work. The cycle is characterized by transfers of heat and work. It is no more a heat cycle than it is a work cycle. Neither heat nor work can execute a cycle. The words are simply names for particular modes of energy transfer from body to body. These transfers may be accounted for by a sort of thermodynamic auditing system, yielding the familiar heat-balance account, but this, like the work of the bookkeeper, is simply a statement telling where something comes from and whither it goes. There is no heat cycle. That which is gone is gone forever.

The science of thermodynamics gives much attention to cyclic processes, and to distinguish such processes from purely mechanical or kinematic cycles executed by trains of mechanism they are commonly referred to as thermodynamic cycles. The term "heat cycle" may perhaps be intended as an abbreviation of this expression, but in any specific case, when the working substance is known, it would appear preferable to designate the cycle by the name of the substance, real or imaginary, as steam, air, mercury, ideal gas, etc., or by the name of the man who first described the cycle, as Carnot, Rankine, Otto, etc., or by the nature of the cycle, as regenerative, etc.

Particularly, in a context where the discussion could not intelligently be misinterpreted, it seems unnecessary to qualify the term at all, but sufficient to say simply "cycle."

Professor Ellenwood has alluded to the statement in Mechanical Engineering that "The main turbine will be bled in order

[1] Professor of Heat-Power Engineering, Sibley College, Cornell University.

to complete the heat cycle." Of course, the exact expression is that the turbine will be bled for the recovery of heat.

C. Harold Berry.[2]

New York, N. Y.

[The expression to which Professor Ellenwood demurs in his letter above appeared in the 1926 report on Progress in Steam-Power Engineering, contributed by the Power Division of the Society. The communications from Professor Ellenwood and Mr. Berry were submitted to H. B. Reynolds, Chairman of the Executive Committee of the Power Division at the time the report was prepared, who replies briefly below.—Editor.]

To the Editor:

Replying to the letters from Professor Ellenwood and Mr. Berry which you have transmitted to me, I would say that my use of the term "heat cycle" was an attempt to get away from the expression "heat balance," which is almost universally used when referring to that phase of power-station design which deals with the flow of energy from the coal pile to the bus bar, together with the various types of apparatus and their arrangement for increasing the thermo efficiency of the power station. I do not feel that the term "heat balance" covers the situation as its use suggests the bookkeeping end of power-station operation.

I think that there is a great need for a brief and descriptive term which power-station engineers can use instead of "heat balance," and Professor Ellenwood and Mr. Berry would be doing the profession a great service if they would suggest such a term.

H. B. Reynolds.[3]

New York, N. Y.

Surface Condensers in Steam Power Plants

To the Editor:

In the paper on surface condensers in steam power plants by J. A. Powell and the writer, published in the May issue of Mechanical Engineering, some of the basic figures used for the calculation of the curves were not given, in order to keep the paper as short as possible. As the study may be found of more value to engineers dealing with the problems of economical condenser installations when the basic figures used for the calculations are given, these will be presented here.

The steam conditions at the inlet nozzle of the turbine were assumed to be 385 lb. per sq. in. gage pressure at a total temperature of 700 deg. fahr. The calculation was based on an overall boiler efficiency of 83.5 per cent, and the coal price assumed to be $4.50 per ton, delivered.

The tubes considered were 21 ft. in effective length, No. 17 B.W.G.

The tube maintenance costs on condensers with more than 20,000 sq. ft. surface were figured at 7 cents per additional square foot.

The cost for auxiliary power was assumed at $40 per hp. per year, this figure including production cost and fixed charges on the proportional part of the station capacity.

A fixed charge of 17 per cent on a capital investment of $3 per additional square foot of surface was figured.

The curves arrived at in the study made refer to normal conditions prevalent in the eastern states. It is evident that these conclusions are subject to revisions according to the local conditions of the plant in question. The general arrangements may favor the single-pass or two-pass condenser, or there may be considerable saving in construction costs of one or another. It may be argued that the single-pass condenser has a greater water-tunnel cost, but this is generally not as much as might be expected and in some cases will be even less than for a two-pass condenser. The local conditions must be given due consideration before a decision regarding the size of the condenser is made.

What the study seems to indicate is the fact that for a plant working under fairly normal conditions the surface of the most economical condenser is smaller than that used in former years. This seems reasonable when it is considered that the following points all tend to reduce the surface of a condenser:

1 More careful operation, better cleaning of tubes
2 Reduction in steam consumption per kilowatt of capacity with the higher pressures and temperatures used
3 Reduction per kilowatt of capacity of steam going to the condenser by the use of bled steam for feedwater heating
4 Improvement in condenser design.

With the high efficiency obtainable with modern boilers and turbines, it would seem that improvements during the next few years will be made mainly along economical lines, instead of in efficiency. This renders it essential to make careful studies regarding the most economical condenser installation, and it is with this in mind that the authors have presented their paper.

H. I. Vetlesen.[4]

Reading, Pa.

[2] Associate Editor, *Power*. Mem. A.S.M.E.

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A.S.M.E. Boiler Code Committee Reports

The Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, Mr. C. W. Obert, 29 West 39th St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply, is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of the Society for approval, after which it is issued to the inquirer and simultaneously published in Mechanical Engineering.

Below are given records of the interpretations of the Committee in Cases Nos. 541, 545, and 546, as formulated at the meeting of March 18, 1927, all having been approved by the Council. In accordance with established practice, names of inquirers have been omitted.

Case No. 541

Inquiry: An interpretation is requested concerning the number of threads required in tapped openings for washout plugs. Will it not be permissible to use a greater number of finer threads for washout plug openings than would be required under Table P-10 of the Code for pipe-thread connections to boilers?

Reply: It is the opinion of the Committee that while washout plugs as specified in Pars. P-266 and P-267 are not permitted to have a lesser number of threads than specified in Table P-10, there is nothing in the Code to prevent the use of a greater number of threads, provided the depth of the threaded hole is sufficient to meet the requirements of Table P-10.

Case No. 545

(In the hands of the Committee)

Case No. 546

Inquiry: Is it the intent of Par. H-26 and Table H-4 of the Code that items *a* and *b* relate only to screwed staybolts, the heads of which may be riveted over or welded to the plate, or do they mean that an unthreaded stay may be welded into the plate and allowed a loading of 7500 lb. per sq. in. of net section? The latter does not appear to be in agreement with item *e*, which allows only 6000 lb. per sq. in. on welded portions of stays.

Reply: Items *a* and *b* of Table H-4 permit either screwed staybolts with heads riveted over, or unthreaded staybolts welded into the plate as specified in Par. H-83, and an allowance of 7500 or 8000 lb. according to whether they are solid or hollow. The weld referred to in item *e* does not apply to the attachment of the stay to the sheet.

Engineering and Industrial Standardization

Refrigeration Code to Be Revised

IN ANNOUNCING a revision of the Refrigeration Safety Code, F. E. Matthews, chairman of the American Engineering Standards Committee's sectional committee in charge of the code, says that in the field of refrigeration, the refrigerating engineer, the natural custodian of safety, first extended his activities from private to public problems upward of a score of years ago when The American Society of Refrigerating Engineers through its code committee made the first rough draft of a refrigeration safety code which, with some elaboration, was finally adopted by city of New York. The interest at that time was purely local.

The movement toward protective law-making rapidly increased, and it became apparent that sound technical advice would have to be pretty generally dispensed if a chaotic condition in the industry was to be avoided. A large standing committee has accordingly been maintained ever since, functioning periodically for the revision of the code as the development of the art and other conditions required.

Between six and seven thousand copies of the code in its various stages of development have been distributed to interested parties throughout the United States, with the result that a number of states and municipalities have adopted the Code Committee's recommendation, in whole or in part, as has best fitted their requirements. The code in its present form was officially adopted by the American Society of Refrigerating Engineers in December, 1925.

Since national scope was soon recognized as necessary for the Refrigeration Safety Code, if it were to fulfil its mission in the enormous and fast-expanding refrigeration industry, and since the American Engineering Standards Committee offered the agency by which this end might be most readily attained, the A.S.R.E. code committee was reorganized, on the formation of the American Engineering Standards Committee, and now functions in accordance with its rules of procedure for the conduct of work which is subsequently to be brought before the committee for approval.

For the revision, the personnel of the sectional committee, which now has 35 members representing 23 organizations, is being reorganized.

The Refrigeration Safety Code deals with the subject of refrigerating equipment under three main divisions made as to capacity. Refrigerating equipment falling in a certain class as to capacity is further classified as to the working medium or refrigerant employed.

Constructive suggestions for the improvement of the code will be particularly appreciated at the present time, and while the code is undergoing revision.

Standardization Within Industrial Companies[1]

THE American Engineering Standards Committee has been collecting material for a considerable period toward the preparation of a series of Sustaining Membership Bulletins dealing with the important question of standardization within the industrial companies. The following extracts are taken from a bulletin recently issued which represents the first instalment of such material. It is intended to serve as a kind of general introduction to subsequent reports that will be more in the nature of analyses of the work of standardization departments in some of the larger industrial companies.

Even in well-coördinated industries, the work of company standardization bureaus is vitally necessary, because in a large organization the interrelationship of shops, tools, etc. makes it necessary to be sure that new standards and changes in standards have been carried into effect in all parts of the plant where they will affect production schedules or alter the character or quality of the product. For example, in an establishment in which the same part may be used in different structures or machines, it is necessary, if a standard is developed or changed, to be sure that the modification has taken into account not merely some of the uses of the part, but all of them, and some inexpensive change in an apparently minor detail may suffice to increase the range of adaptation of the part in an important way.

Moreover such standards must be carefully compared with corresponding elements of national and trade association standards, in order to avoid any important inconsistency, want of interchangeability, or difficulty of procurement that might arise.

A number of American companies have done standardization work for some time and a few have developed quite elaborate standards departments, some of which publish comprehensive printed or blueprinted standards which are well known in their industries. The General Motors Corporation is prominent among these, and a number of other automobile manufacturers have given continued and careful attention to this subject. The American Engineering Standards Committee is indebted for information among others, to The Stewart-Warner Speedometer Corporation, The General Motors Corporation, The Allis-Chalmers Manufacturing Company, The Studebaker Corporation, The Republic Flow Meters Company, Westinghouse Electric and Manufacturing Company, The Crompton and Knowles Loom Works, The Singer Manufacturing Company, The Dexter Folder Company, The Foxboro Company, and The Gleason Works. (Information on the drawing-room standards of the various companies which kindly furnished material in this field, has already been presented in Sustaining Membership Bulletin 11.) The A.E.S.C. will be greatly indebted to other firms who will make their material available for study and so facilitate the preparation of this compilation of company standardization practice in the United States, which it is hoped may prove of considerable general interest.

The question often arises whether new standards should be gradually developed in the proper department and then instantly put into effect as of a certain date, by official order of the management, or whether the standards should be fed out into the industry as it were, as fast as they are developed. If the industry is a close-knit one and the standardization is part of a radical revision of the management methods, for which the individual executives are prepared and with which they are in sympathy, then the first method may be feasible; but in general, in well-established firms the effect of traditional attitudes of the foreman, workmen, etc. is such as to make the gradual method of introduction more satisfactory, and certainly far easier from the point of view of human relationships. Moreover it permits the work done to be corrected by the experience accumulated as standards are introduced, and so prevents what may occasionally be very serious and far-reaching errors of judgment.

The use of national standards within company organizations ordinarily involves some interpretations or condensation, and naturally a highly generalized standard meant to cover the needs of a whole range of industries, e.g., a standard for bolts and nuts, for screw threads, or for shafting, cannot be completely adapted in this general form to the particular needs of a single company. It may call for too large a number of types and sizes, or it may not provide for some specially good or specially poor quality of product needed for a particular work. Therefore the practice usually is to reproduce the national or trade association, or technical society, standard in some convenient and uniform style, perhaps in a loose-leaf handbook, or on charts to be tacked to the wall, in which all of the company's own practices with regard to the elimination of intermediate sizes and the like are crystallized. In Germany it has been the custom to use the national standard

[1] The material for these notes was taken from a recent bulletin issued by the American Engineering Standards Committee to its sustaining members. This Committee, whose offices are at 29 West 39th Street, New York, N. Y., would be interested in receiving comments on the various phases of this subject from the readers of MECHANICAL ENGINEERING.

sheets very extensively, with a special mode of indication of regular stock sizes, sizes that can be furnished when specially called for, sizes that are never used, etc., directly imprinted upon the standard sheet as supplied by the national standardizing body. This has the advantage of saving considerable in expense, but is not well adapted to the form of publication which has up to now been most common in the United States. It is particularly suited to loose-leaf, single-page presentation of standards, which is the regular method in use on the Continent. For instance, the Germans, instead of having a pamphlet covering screw threads of all types and sizes, will have a large group of single sheets. The first sheet usually gives a general view or survey in convenient form, of the whole field, showing the scheme of gradation of sizes, the different classes, if any, specific references to detailed sheets, etc. Another sheet may show Whitworth fine thread, type 1, from 56 to 499 mm. diameter. Another may show single trapezoidal threads from 10 to 300 mm. diameter, and so on. Thus, the sheets are split into convenient small units, suitable for the use, if necessary, of the individual mechanic or tool setter who may be working on a single standard type of screw or machine element for a long period of time.

On the company standards committee, representation of the drafting and design departments, the production department, purchasing, stores, and sales departments, and shop superintendents or foremen and perhaps others, will need to be provided for. If any one of these important groups is omitted from consideration in the standardization work, there may be large and unexpected resistance to the acceptance of the standard when it is ready, or important factors that had been neglected may be brought to light when correction is difficult and expensive.

Every facility should be afforded the chief of the standards department, and perhaps some members of his staff, to participate in national technical and trade association standardization work. The contract so made will be of inestimable value in bringing to the plant information about how other firms have met similar problems, and furthermore will keep the organization in close touch with new developments in industry that affect standards through changes in processes of manufacture and of assembly, new materials available for production, etc. As one man has put it, such contacts enable people to get in on the ground floor with reference to ideas that are still new and have important industrial reflexes.

An important decision that must be made in each firm is whether the standards work should be an adjunct of the designing department, or whether it should be connected more directly to the higher management of the organization. The practice in Germany seems distinctly to favor the latter arrangement, and the standards departments there almost uniformly report to the general superintendent of the plant. The amount of authority which the standards department should have is again a matter of company policy. From the human point of view, it seems advantageous to put large authority in the departments which develop the standards under the general direction of the standards departments, while keeping the standards bureau itself in a sort of advisory and correlating relationship. If the proper departments are represented on the various standards committees, there will be no difficulty in insuring prompt realization of the work that is done. In one point, however, the practice seems common of requiring that use of special parts or materials not authorized by the standards shall be validated by some high administrative official, such as the chief of the standards department, an assistant to the general superintendent, or the like. It is valuable, in cases where not too much inconvenience is involved, to route copies of all drawings for new designs through the standards department, in order that they may be effective as suggestions for new fields in which standards may be needed, or for modifications in old standards that may make them more widely adaptable or cheaper to manufacture.

The standards department should be provided with excellent facilities for the economical duplication of drawings and specifications. In general, it will pay to circulate drafts of proposed standards rather more widely than would ordinarily be thought necessary. The extra interest and attention brought to bear by this method often reduces quite unconsciously the resistance to the introduction of new standards when the draft is finally worked over and completed. Anything which tends to eliminate the feeling of complete innovation will favor the ready adoption of standards. In the case of one company, a proposed new standard went through ten departments before final agreement was reached. Such work is naturally slow and difficult, and those engaged in the work of the standards department must be chosen with a very definite view to their patience in working out complicated and interrelated problems requiring frequent retracing of steps and replanning of details and outlines. The slow and rather expensive process implied herein is nevertheless so effective in bringing about savings that little thought needs to be given to the cost of the work in all cases where production is large and intricate. Even very small savings on a single item, which are almost sure to be produced by the general process of cancelling out of duplications and adjustment of minor and non-essential differences, which the standardization work calls for, will be great in the aggregate, and ordinarily sufficient to cover the cost of the standardization work many times over. It has not been easy to collect instances of losses due to want of standardization in individual companies, and by the same token the monetary savings occasioned by the introduction of standards are difficult to define. However, the following incident, although relating to a small item, may perhaps be regarded as typical of the wastes which occur in manufacture in large organizations when there is no standards department or equivalent means of coördinating the design and production of components for machinery. In one American machine tool plant recently it was found that there were *five cylindrical pins of one size* made in connection with different jobs, stored in separate bins and no record of similarity to be had. Two of these pins were practically obsolete, two others were produced in small quantities, and only one of them was made efficiently in large quantities on an automatic screw machine. The cost of the pins produced in small quantity was 400 per cent of the cost of pins produced by automatic machines in large quantities. Besides these five pins which were identical in design, there were five or ten more pins sufficiently like the five mentioned that they could have been reduced to the same design without loss of effectiveness, and would thus have resulted in economies involved in reduced tool cost, storage space, recording and bookkeeping, inventory, mental effort decreased obsolescence, greater turnover, and simplification of the problem of design of machines using these components.

In the same firm it was found that there were about 1000 varieties of miter gears designed for driving two shafts at right angles at a 1:1 speed ratio. The savings that will be made by redesigning, classifying, and standardizing these miter gears on a minimum number of sizes and stresses are obvious enough to need no detailed discussion.

Referring to S. M. Bulletin 10, page 11, the very large reduction occasioned by standardization in the stock of one company on typical manufacturing items such as nuts, washers, collars, pins, etc., is shown to range from 20 to 56 per cent.

The nature and magnitude of the savings to be made in mass production of electric motors under standardization are outlined in an abstract of an article in a British journal (S. M. Bulletin 12, item 26, page 8). An English firm asserts in a full-page advertisement in a recent number of the *Electrical Review* that it has been able to produce a saving in price of at least 50 per cent below other manufacturers on alternating-current motors of superior quality and efficiency, which are completely standardized and produced in a factory specializing in that one article.

It will be desirable to keep track of a few of the more striking savings produced through the standardization work for the sake of illustration and for use in discussions with those executives and workmen throughout the plant who might be inclined to doubt the value of the procedure, or of certain aspects of it.

The extent to which Sweden's manufacture of ball and roller bearings has been expanded since the outbreak of the World War is indicated in the fact that, according to Swedish customs returns, the value of these exports in 1926 represented more than six times the value of the exports of 1913. The actual figures for these years were as follows: 1913, \$978,063; 1926, \$6,239,062. The requirements of the automobile industry in the United States are stated to have created an increased demand for Swedish ball bearings.

Plant-Location Factors of Western Electric Co., Kearny Works

By O. C. SPURLING,[1] NEW YORK, N. Y.

In 1922 the lead-covered-cable department at the Hawthorne Works of the Western Electric Company was working overtime and a regular night shift besides, in order to meet the needs of the Bell System, and as forecasts of future requirements indicated even larger demands, it was decided to add to the cable manufacturing facilities either at Hawthorne or elsewhere.

All other things being equal, an industry generally locates near its customers, but the location of the sources of raw materials is always an important factor and the site chosen is usually a compromise because of transporation factors.

The cable department requires semi-skilled and skilled labor and it takes time to build up an industrial community of such workers. As a rule they do not wish to leave their environment without a considerable increase in wages or other compensations. Even when they do migrate they easily become discontented and return to their old surroundings.

Cities, being natural centers for trunk lines or water transportation, usually offer superior advantages for obtaining raw materials and for shipping finished goods. An abundant labor supply is obtainable as compared to other locations. A plant located in a city generally enjoys municipal advantages such as good streets, gas, sewers, police, and fire protection, etc.

Some of the disadvantages of a city location are—

1 Land is high-priced.

2 It is often difficult for a large works to secure a site within a city where buildings exactly suited to the purpose can be erected without great expense.

3 If the city is a growing one, taxes in time make the location expensive.

4 City restrictions and municipal regulations have to be carefully considered.

5 While labor may be abundant in the city, the cost of living and hence the wages paid, are in general higher than in the country.

In considering our particular problem, comparisons were made for different general locations, of combined costs of raw materials, and of manufacturing processes and transportation of both raw materials and finished product. Studies of relative freight rates where rail and water transportation were in competition with each other showed so little difference between the two methods that we assumed all transportation charges would be in accordance with rates by rail.

The sources of our principal raw materials were as follows: Lead from Missouri; paper from Troy and Boston; antimony from New York; copper from Hastings, Baltimore, Bayway, Chicago, and Kenosha.

Our finished product was distributed as follows: 45 per cent to the states bordering on the Atlantic Ocean and the Gulf of Mexico, east of the Mississippi River; 45 per cent to the central and central-west states; 10 per cent to the mountain and Pacific states.

The lowest combined total costs for the additional manufacturing facilities were found to be for the general location of New York City.

Hawthorne had the lowest cost of raw materials, including transportation charges, and lowest distributing costs of the finished cable for the central, central-west, mountain, and northern Pacific states.

For the Atlantic, Gulf, and southern Pacific states, a site in Virginia compared with Hawthorne saved approximately $200,000 a year in distributing costs, but that was balanced by an increase of the same amount in cost of raw materials.

A similar comparison for the Kearny site showed a saving of approximately $400,000 a year in distributing costs and an increase of $125,000 a year of raw materials, or a net saving over Hawthorne for the territory mentioned of $275,000 a year.

The primary object was to find a plot of land of not less than fifteen acres to accommodate 200,000 sq. ft. of one-story manufacturing space and large areas for storage of empty and loaded cable reels. In order to have some margin for growth, twenty-five acres was considered desirable. The choice locations in the Metropolitan district had been acquired and developed many years before, so that we had only the leavings from which to choose.

We required—(1) First-class facilities for receiving and shipping by rail, by roadway, and by water; (2) an adequate supply of labor; (3) water supply; and (4) sewer system, etc.

Real-estate agents were interviewed and a tug was chartered to tour the New York upper bay district, the Arthur Kill, the Kill von Kull, Newark Bay, and the Passaic and Hackensack rivers. Forty-two properties were considered. Five were on Long Island, 2 in the Bronx, 3 on Staten Island, and 32 in New Jersey, extending from Paterson on the north to Perth Amboy on the south. The locations on Long Island and the Bronx were rejected because of the need for using the East River for water transportation. This river is dangerous to navigation, especially in the Hell Gate section. On Long Island there was the further objection of congested freight-transportation facilities. On Staten Island the locations available were in the extreme southwest section where railroad facilities were not available, and the supply of labor was not considered sufficient to serve our needs.

Most of the New Jersey sites had good rail transportation, but there was considerable variation in water facilities and supply of suitable labor. Some of the properties were rather small in area. After careful consideration, it appeared that the Kearny site more closely approximated the important requisites than any of the others, and after boring and piling tests had been conducted in order to give careful consideration both from an engineering and financial viewpoint, the Kearny site was chosen and purchased in February, 1923. It fronts on the Passaic River and lies between the New York & Newark branch of the Central Railroad of New Jersey and the Lincoln Highway.

The Kearny site had the following advantages:

1 First class facilities for transportation by rail and water

2 Convenient location for local and long-distance shipment by automobile trucks, including accessibility to the new Holland tunnel.

3 Adequate labor supply—one and three-quarter million people having their homes within 15 miles of the property, that is, within an hour's ride.

4 Convenient transportation for employees by the New York & Newark branch of the Central Railroad of New Jersey.

5 Nearness to Lincoln Highway, with street-car and bus service to adjoining cities.

6 The surface of the ground was approximately level, eliminating special grading operations.

The disadvantages were:

1 About one-fourth of the property was under water, calling for bringing in large quantities of filling material.

2 The soil was marshy, necessitating pile foundations under all of the building-construction work.

The site contained fifty-five acres and was considerably larger than necessary for the development of the original cable-plant problem, but it was necessary to purchase the entire tract, and in the preparation of a plot plan to develop the entire area, it was decided to provide for some multi-story buildings for possible expansion in the manufacture of telephone-central-office equipment. This proved to be a wise decision because that expansion had to be provided for before the cable-plant buildings were ready for use.

The Kearny Works is now producing about 40 per cent of the company's total output of cable, serving the states bordering on the Atlantic Ocean and the Gulf of Mexico east of the Mississippi River, and southern California territory around Los Angeles by way of the Panama Canal. About 700,000 sq. ft. of multi-story buildings are in service and the ultimate development of the property provides for 3,000,000 sq. ft. of floor space, requiring 25,000 to 30,000 employees to properly man it.

[1] Plant Engineer, Western Electric Co., Inc. Mem. A.S.M.E.

Contributed by the Management Division and presented at a meeting of the Metropolitan Section of the A.S.M.E., New York, April 13, 1927.

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and Attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities, Papers and Proceedings of

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A Notable Anniversary

OUR contemporary and friend, the *American Machinist*, has attained to its fiftieth birthday, and celebrates the event by publishing, under the date of May 19, a most admirable anniversary number.

The text pages lead off with a message—"Fifty Years"—by our president, Mr. Schwab. This is followed by a series of vivid, condensed résumés of progress by competent and authoritative observers in all fields allied to the machine-tool industry. The list includes: the Hon. Wm. Jardine, Secretary of Agriculture; Gerard Swope, president of the General Electric Co., President Truesdale of the Delaware & Hudson Railway; Vernon Kellogg of the Academy of Sciences; Secretary Hoover, and many others.

A further section is devoted to textual and pictorial developments in particular branches of the machine-tool field, followed by a group of one-page articles by specialists in the fundamental divisions of shop operation and management.

Altogether this number is the most interesting and effective example of technical journalism that we can call to mind.

Our own Society is approaching its half-century mark. It is no mere fortuitous circumstance that makes The American Society of Mechanical Engineers and the *American Machinist* so nearly of an age. Though first conceived (so far as we know) in the brain of Professor Sweet, the Society was born in the editorial rooms of our contemporary under the kindly and fostering hospitality of its first publisher, Horace B. Miller. Without Mr. Miller's tact and ability it is doubtful if there would have been any early resolution of the various differences and degrees of opinion among the founders.

Many thanks for old services rendered and friendly relations preserved! Best wishes for future years of comity and service!

Saving at the Spigot and Wasting at the Bung

THE New England Council is doing a highly commendable piece of work in driving home to manufacturers in the states which they serve the vital effect of manufacturing equipment upon industrial prosperity. While their studies thus far have been largely concerned with the textile and shoe industries, the conditions uncovered are found to apply with equal or even greater force in the metal-working field.

There are many traditional industries which up to very recently were almost preëminent, but which now find themselves facing competition threatening not only their preëminence but their very existence. Such industries are by no means confined to New England—they will be found in any of the older industrial regions. There may be several reasons for their condition, but almost invariably the major reason is that through a false sense of economy they are attempting to turn out a competitive product by the use of tools of ancient vintage.

Machine tools do not necessarily wear out in five, ten, twenty, or even fifty years—although it is safe to say that those of the high-production variety have seen their best days in ten years if they have been pushed to capacity over that period. What most certainly does happen, however, is that in five, ten, or in fifteen years at the most, progress passes machine tools by, and whatever may be their physical condition at the end of that time, they are definitely obsolete nevertheless.

Accountants are quite well agreed upon the advisability of "writing off" the machine-tool equipment of production shops within a period of ten years. There has been, unfortunately, by no means as clear an understanding on the part of those who hold the purse strings that machine tools have a definite retirement age in fact as well as upon the books.

When this realization is driven home, either by further failures among the older industries or preferably by the warnings of such organizations as the New England Council, a part of the surplus of many concerns will promptly be invested in substantial new machine-tool equipment. These concerns will thereby escape the otherwise inevitable day of reckoning when all of the surplus fades into thin air in a hopeless effort to cover up operating deficits which persistently refuse to remain covered.

The Mechanical Engineer and Our Country's Economic Progress

THE wealth and economic progress of the United States have no parallel in history. The richest countries of the past had probably a total wealth far below what several states of the Union can show today. The total wealth of the United States is not only stupendous in itself, but it is growing at a rate which would have been considered absolutely unbelievable only a score of years ago. In the last decade the wealth of the United States has almost doubled and today is estimated at more than \$300,000,000,000. The national income is estimated at \$89,000,000,000, which is greater than the total national wealth of Great Britain. The daily revenue of American railroads is close to \$20,000,000. The bank clearings are of the order of \$1,500,000,000 a day, while 7,000,000 telegraph messages and 72,000,000 telephones calls are required every twenty-four hours to handle the vast business of the country.

When one tries, however, to determine the vital factors which led to this great economic development of the country, he is struck particularly by the fact that the development of the territory and industries of the United States was accomplished with an amazingly small amount of man power.

Here was a continent stretching some 2000 miles from the Canadian frontier to the Mexican border and more than 3000 miles from the Atlantic to the Pacific. It was not an easy continent to develop, intersected as it was by primeval forests, great and often unruly rivers, many mountain ranges, and deserts, and to make things still more difficult, a warlike and unfriendly aboriginal population.

Under somewhat similar conditions in Australia and Canada development has been slow and is still limited to comparatively narrow belts along rivers and along the few existing railroad lines. In the United States it covers practically every foot of the territory.

It is obvious that the few white pioneers could not have carried through these great developments had they not been assisted by the flood of mechanical devices which were made available from the very beginning of American civilization by native talent.

While railroads were first built in England, the rail mileage in America nevertheless soon exceeded that of their native country.

Agricultural machinery was applied on a large scale in America long before Europe found any use for it. Later all kinds of labor-saving devices were developed here and found a ready market, while, either through tradition or conservatism of business methods, Europe looked on them with suspicion and often aversion. This was the case with the sewing machine, typewriter, cash register, and, later, the automobile. Labor-saving devices in the homes, such as vacuum cleaners, washing machines, electric ironers, so common in American households today, are still barely known in Europe, while the mechanized American farm with its tractor and many power-operated devices, electric light in the barns and farmhouses, electric milking machines, and cream separators and a motor truck to take the various products to market, represents an ideal which European farming does not expect to reach for many years.

To whatever branch of American activities we turn, we always find the underlying idea of employing man power not to do the actual work but merely to supervise and direct the work of machinery. The American industrial army has few human privates, mechanical slaves doing most of the work and man merely ordering them about, which is the secret of the country's ability to produce in many lines goods at the cheapest prices with the most highly paid labor in the world.

If we examine the means by which the actual work of the country is done, from the 200,000-hp. turbo-generator—which in Europe would satisfy all the demands for electric current of some ten independent countries that could be named—to the little suction fan in the vacuum cleaner with which the American housewife keeps her carpets and furniture clean, we find mechanical devices underlying it all. Substantially, it is the efforts of American mechanical engineers, taking this term in its broad sense, which have created the foundation underlying the great prosperity of this country—a fact all the more remarkable in that the mechanical engineers themselves in the stress and hurry of the work of the day are apt to forget about it. And yet it is true, and it is useful to realize that it is true, for it indicates the vast field of future possibilities still open to American mechanical engineering.

The Mechanical Engineer in Agriculture

With the advent of agriculture came various forms of crude implements for digging and planting in the fertile earth of the tropical regions where the major portion of the earth's population then existed. Evidently these implements were made from sharpened sticks and bones, and formed the first period of development. As demands for space made it necessary that some of the people reside on the uplands and in the more temperate zones, there came a second period when deeper cultivation was necessary, which resulted in the development of crude plows drawn by beasts of burden. A third period followed many years later when implements of metal were introduced for tillage of lands still more difficult to cultivate profitably. We are now in the early part of a fourth period, that of machinery and power application.

Prior to about 80 years ago all agricultural implements were fashioned in blacksmith shops, or even by the farmer himself; consequently there was little uniformity of design, although engineering effort had long since been turned to good advantage upon other problems affecting man's existence. This is not difficult to understand in some of the European countries where class distinctions are sharply drawn, for no doubt it is difficult for one of high standing to see any advantage in studying the implements of the lowly peasant.

In our own country, however, these class distinctions are not so sharply drawn, yet it is only comparatively recently that the mechanical engineer has come to devote really serious attention to the farmer and his efforts to produce greater crops per man power.

A paper by O. B. Zimmerman, presented before the Kansas City meeting of the Society, April 4 to 6, 1927, proved most enlightening in this connection. According to the author, the use of improved types of equipment has released from the farms something like 20 million men for other forms of work, yet the American farmer feeds and clothes himself better than the farmer of any other country in the world, and in addition exports great quantities of farm products.

One of the most important factors in this new power era of agriculture is the tractor. For instance, it is now possible for two men to harvest and thresh a field of grain with no more effort than formerly was required by three men and four horses for the cutting operations, and a large and cumbersome machine with a force of about ten men for threshing. It is not difficult to visualize the saving resulting from the elimination of the usual cutting, shocking, hauling, and stacking operations, and possible spoilage accompanying periods of waiting for the community threshing outfit. Crops are cultivated four times as rapidly as formerly, and with one-third to one-quarter the manual labor. New types of tractors with frame clearances permitting straddling of waist-high rows of plants carry many of the implements formerly drawn by horses. Seeders of twice the capacity of the horse-drawn type are drawn with almost twice their speed, and nightfall finds a continuation of operations with lighting equipment.

The eternal battle against insect pests is being more successfully waged with the assistance of mechanical equipment. Dusting and spraying operations in the cotton, potato, and tobacco fields are handled directly from the tractor, and even airplanes are pressed into service in large fields and orchards as distributers of poison. Even the corn borer, one of the most persistent pests with which the farmer has to contend today, finds a properly mechanized farm a most unhealthy place.

These are but a few of the accomplishments of the mechanical engineer after his belated entry into this very important field. What may be accomplished in the future, after the needs are more clearly understood, and with the farmer's son no longer interested in old Dobbin and the "one-hoss" plow, but things mechanical and the state university, is indeed something to appeal to the imagination.

The Mississippi Rages

The Mississippi Valley is in the midst of a great calamity, the worst in history. Lives have been lost, great areas flooded, thousands of houses destroyed, and the orderly processes of agriculture and commerce deranged. Modern engineering skill is challenged by this latest onslaught of the Father of Waters rolling on in irresistible flood.

The Mississippi presents problems on a scale found nowhere else in the civilized world. It surpasses in flood volume all other rivers. It flows over beds of soft alluvial deposits which readily erode. Its sediment is almost impalpable. This combination of factors is not found in any other river which has been studied. It is therefore not possible to apply to the Mississippi the results obtained in the control of any other river.

The control of the Mississippi River is too complicated a problem to be discussed in this brief space. It lies outside the technical range of the mechanical engineer, but as a public question of far-reaching importance it affects industry throughout the country, and mechanical engineers have their place in its discussion. A long view ahead reveals possibilities for the greater use of the Mississippi for navigation and for the greater reclamation of swamp lands.

Relief efforts are the order of the day, but much newspaper space is being given to a discussion of the immediate steps which should be taken to reduce the dangerous effects of high floods in this mighty river. Various plans have been advanced. Reforestation, building of reservoirs on headwaters, the construction of detention reservoirs, and the locating of permanent controlled spillway sluices are advocated. There is clamor for the calling of a session of Congress to appropriate huge sums of money for immediately building great works. In this excitement, one calm note is heard. The American Engineering Council passed a resolution urging the appointment of an engineering commission to examine into and report upon the broad problem of a comprehensive plan for the control of the Mississippi River and the prevention of a repetition of the disastrous conditions now existing in consequence of the recent floods. The Council also urged that the Ransdell Bill for the establishment of a hydraulic laboratory be reintroduced into Congress and pressed for early passage.

Of the many procedures which should be followed in finding an improved method of curbing the Mississippi and its tributaries, the hydraulic laboratory offers one of great importance. This type of laboratory, which has been developed extensively abroad during the last twenty years, permits the erection of scale models of flow conditions in which various elements of the problem can be separated for study and some fundamental principles secured for guidance in applying corrective measures.

The Mississippi Valley is heavily populated, it has great wealth, and it has made great contributions to the welfare of the nation. Its problems are the problems of the entire country. Engineers throughout the land are interested in stabilizing the flow conditions of this great body of water. As citizens they can counsel fact-finding measures to secure greater knowledge of the problems to be solved, and as engineers they extend to their confrères in the field of civil engineering the support and the help they need in their great work.

Aeronautic Division Holds Its First National Meeting

The first National Meeting of the Aeronautic Division of the Society was brought to a close on Tuesday afternoon, April 26, after a most successful two-day convention in Buffalo, N. Y.

The meeting opened Monday morning at ten o'clock with a few words of greeting from A. H. Lane, chairman of the Buffalo Section, and a short address of welcome by Mayor Schwab of Buffalo, who presented the keys of the city to the visitors. The presiding officer of the session, C. Roy Keys, then introduced Anthony Fokker, who presented his paper on Transport Airplanes. The second paper of the morning session, presented by Harry F. Guggenheim, was on the Importance of Aerodynamic Safety on Aviation. This paper dealt especially with the technical tests of the New Samuel Guggenheim Safe Aircraft Competition.

Following the morning session there were interesting inspection trips to the Curtiss Aeroplane & Motor Plant and to the Consolidated Aircraft Plant.

At the banquet on Monday evening, H. Ralph Badger, aviation governor of New York State in the National Aeronautic Association, the toastmaster, introduced Elmer A. Sperry, 1927 John Fritz Medalist, who represented President Schwab and delivered the following message from him: "Air transportation will develop all the relations between the people of one nation and between all nations to an even greater extent than did the railways. I wish you would announce for the Society that it will put its energies into developing this science both by research and special meetings and particularly by special publications."

Mr. Sperry spoke on the Necessity of Developing a Heavy-Fuel-Oil Aero Engine. The Rev. G. A. Leichliter of Buffalo gave an amusing talk entitled Lighter than Air. The last speaker was William P. McCracken, assistant secretary of Commerce in charge of aviation, who spoke on Commercial Air Transportation.

Three papers were presented at the final technical session on Tuesday morning, all calling forth discussion. Earl D. Osborne of New York presided, and the first paper was presented by Col. V. E. Clark, vice-president of the Consolidated Aircraft Corporation, on Apparent Present Tendencies in Airplane Design. The second paper was by Bishop Clements, on Metallurgy of Aircraft Engines. Major John M. Satterfield closed the session with his paper on Municipal Airports.

On Tuesday afternoon a trip was made to the Airport, where the airways were in perfect condition. The large Ford all-metal airplane was held for inspection before it left on its return trip to Detroit, and an opportunity was given to every one to examine it inside and out. The plane was loaded with 1403 lb. of freight, and its take-off was perfect and a sight worth remembering. Many different types of planes parked in the hangars were also seen at the Airport. Lieut. Reuben Biggs and Lieut. E. E. Aldrin, chairman of the Aeronautic Division, gave an exhibition of stunt flying, using a "Trusty" preliminary training plane of the Army.

From the Airport the visitors were taken to the plant of the Eberhart Aeroplane & Motor Corporation where they were afforded an opportunity to inspect a Navy battleship fighter plane that was in the process of construction.

The Daniel Guggenheim Safe Aircraft Competition

The final rules for this competition were announced at a dinner at the Yale Club, New York City, on April 29. Its purpose, of course, is primarily to assist in the development of aircraft safer than these generally available today. It is expected, however, by its promoters that the discussions and publicity which will result from such a competition will help to instill in the general public greater confidence even in the aircraft types existing today.

The applications for entry will be received on and after September 1, 1927. An entrance guarantee of $100 must be forwarded with the application and will be returned upon rejection of the entry or upon the acceptance and presentation of the aircraft for test. Certain information regarding the aircraft must be also submitted at the time of entry, but no restriction is made as to the design or country of origin of the aircraft, provided it is a heavier-than-air machine. The competition is to close on October 31, 1929, but the Fund may advance the date of the closure if it so desires.

From an engineering point of view, the greatest interest attaches to the qualifying requirements, in particular those of performance and load. In order to be acceptable for the competition the aircraft must satisfy the following minimum requirements in regard to performance: maximum speed, 110 m.p.h., rate of climb, 400 ft. per min., and a useful load consisting of 5 lb. per hp., which shall include pilot, observer, fuel, oil, and instruments. In the tests themselves the aircraft shall maintain level and controlled flight at a speed not in excess of 35 m.p.h. and shall be able to glide for a period of 3 min. with all power switched off, during which time the air speed shall never exceed 38 m.p.h.

The aircraft shall land with all power switched off, and after first touching the ground shall come to rest within a distance of 100 ft. (Braking devices shall be permitted with certain limitations.)

The aircraft shall make a steady glide in over an obstruction 35 ft. high and land in a straight line with all power switched off. After landing, the aircraft shall come to rest within a distance of 300 ft. from the base of obstruction. The approach of the landing ground shall be straight; turning, side slipping, or trick flying will not be permitted.

The aircraft shall take off after running not more than 300 ft. from a standing start. The aircraft will not be considered to have passed this test if it touches the ground again after taking off.

After taking off within a distance of 300 ft. from a standing start, the aircraft shall clear an obstruction 35 ft. high at a distance of 500 ft. from rest. The approach to the obstruction shall be straight, and trick flying will not be permitted.

Longitudinal Stability. The aircraft shall be provided with means by which it can be trimmed so as to fly with the elevator control free at any speed within the range of 45 m.p.h. to 100 m.p.h. and at any throttle opening of the engine or engines. The test of longitudinal stability shall be as follows:

The elevator control to be moved toward its maximum extent either backward or forward sufficiently to give a fair test of stability and then released. In either of these cases the aircraft must return to steady flight in its original attitude within a reasonable time.

General Stability. The aircraft to be capable of flying at any air speed from 45 m.p.h. to 100 m.p.h. and at any throttle opening of the engine or engines with all controls left free for a period of not less than 5 min. in gusty air.

Certain tests are also provided to show the ability of aircraft to recover from abnormal conditions and show its controllability and maneuverability both in restricted territory and on the ground.

There can scarcely be any doubt that any aircraft that would satisfy the requirements laid down in the conditions for the competition would represent a very great advance over anything now available, and the very fact that such requirements are already considered sufficiently practical to be presented by an organization of the standing of the Daniel Guggenheim Fund, speaks volumes for the state of development of aeronautical engineering of today.

Annual Meeting of the American Welding Society

THE Annual Meeting of The American Welding Society was held in the Engineering Societies Building in New York, April 27 to 29, 1927, and comprised in addition to the usual social functions, meetings of the various committees and the reading and discussion of technical papers. Among these some were published in the April issue of The Journal of the society and will be briefly reported here. It is expected that the other papers will be published in the May issue, in which case they will be reported in MECHANICAL ENGINEERING if space is available.

The subject of fatigue of welds was discussed by R. R. Moore, who reported on tension and fatigue tests made on various types of specimens, with special attention to structures that might be used in airplane work. (The author is connected with the McCook Field.) He made tests on various kinds of gas and electric welds and among other things found that the endurance limit in gas welds is only 28 per cent of the tensile strength of the welded joint at best. The vanadium filler rod did not improve the endurance limit, but actually gave a lower value. The Norway-iron filler rod gave an endurance limit of only 13,000 lb. per sq. in. as compared with 16,000 lb. for low-carbon-steel filler rod. Specimens welded with chromium-molybdenum filler rods gave a still lower endurance limit (8000 lb.), which was, however, apparently due not to the type of filler rod but to poor welding, as evidenced by lack of penetration of the weld.

The atomic-hydrogen process of welding softened the tube more than either the gas or arc methods, but the endurance limit was approximately the same as in the best gas weld. The author claims to have definitely found that the deposited metal does have an endurance limit just as wrought ferrous metals do.

J. F. Lincoln, vice-president of the Lincoln Electric Co., told why and how certain classes of castings can be replaced by welded steel parts, and mentioned such instances as the rotor spider for an electric motor, the end ring for a motor, the motor base rail, etc. When a 12-in. pulley has been replaced by a welded steel section, it was found that the former weighed 79 lb. and cost \$10.89, and that the latter weighed 40 lb. and cost \$3.05.

Overhead is not included in these costs. The reason this is not done is not because of the fact that it will change the final result, but because of the fact that overhead is a widely varying quantity, and overhead in its amount is, to a considerable extent, controlled by the amount of inventory, as inventory, depreciation, storage, and interest charges are invariably a part of the overhead.

It is self-evident that in general the labor cost is higher with arc-welded steel replacing cast iron than with the original casting. Therefore, if overhead were included with its usual percentage, the ratio would be more favorable to cast iron, although still the arc-welded steel would be very much cheaper. However, the arc-welded steel has one very great advantage over cast iron and that is that the inventory cost necessary for a certain production will be enormously decreased. Therefore one item of inventory will be decreased by this substitution. In addition, the item of patterns, pattern storage, and repair will be entirely eliminated, which also will very greatly affect inventory.

In general it is probably safe to say that the items which would reduce overhead would be largely balanced by the increase necessary because of the increased labor cost, so that the total actual figure, including overhead, in both cases would probably, as an average, balance. This is an estimate only, but indicates the experience that the company with which the author is identified has found to be true.

Much work must still be done in getting the best shapes and the best method of applying these shapes to the work in hand. In the case of buildings there is still much work to be done in the design of the joint for most efficient application of the new process, also for the methods of erection, which become a very serious problem in arriving at the most economical method possible.

Among other things, the author makes a claim that pipe lines in the future will be welded instead of cast.

The Westinghouse Electric & Manufacturing Co. (paper by A. M. Candy, General Engineer) reported on an investigation of structural-steel welding having for its purpose to determine the reliability of this process. The first tests consisted of simple bracket and beam connections, which were followed by subjecting welded structures to vibration and shock tests, both of which proved satisfactory.

The Research Department of the Westinghouse Co. has also carried out a series of tests to show the change in physical strength of weld metal at high temperatures. These showed that while the tensile strength is reduced 25 per cent, the elongation is increased over 50 per cent, and the reduction in area over 50 per cent when the weld metal is heated to a temperature of over 850 deg. fahr.

A group of interesting papers were presented in a symposium of research activities in the welding field during the past three years, some companies describing their equipment and others reporting on the actual research problems handled and results obtained. Only a few of these papers can be reported here.

D. H. Deyoe (Industrial Engineering Dept., General Electric Co.) described a new magnetic-clutch-type automatic welder and control, the essential feature of the former being that the motor revolves at a constant speed in one direction, merely turning the horizontal shaft which carries the two beveled magnets. This constant speed can be adjusted to various values, but once set for a particular job it remains constant. It is claimed that a faster, stronger, and better weld can be made on $1/2$-in. plate or thicker by depositing the welded metal in two or more layers using around 350 amperes in each arc, rather than trying to deposit the metal in one layer using an excessively large amperage, and the multiple-arc welder was developed with this in mind.

A number of practical problems were investigated by the Newport News Shipbuilding and Drydock Co. (paper by Jas. W. Owens, director of welding of the company). Among others the following problems were investigated.

Spot Welding Galvanized Iron. It was found that spot welding of No. 24 to No. 11 gage galvanized sheet iron and steel is very satisfactory, if the stated requirements are observed.

Flash Welding of Tungsten Steel Tips to Medium-Carbon Steel Shanks in the Manufacture of Lathe and Planer Tools. Pieces of tungsten steel $5/8$ in. $\times$ $1\,1/4$ in. $\times$ 2 in. long, $7/8$ in. $\times$ $1\,3/4$ in. $\times$ 3 in. long, $1\,1/4$ in. $\times$ 2 in. $\times$ 4 in. long, and $2\,1/4$ $\times$ $2\,1/4$ $\times$ 4 in. long were flash welded to medium-carbon-steel shanks having a corresponding sectional area, the tools forged, tempered, and service tests made. This method of manufacturing lathe and planer tools and the salvaging of broken tools can be readily performed on the company's present equipment, the cost of manufacturing such tools being approximately only one-third that of tools made of solid tungsten steel.

Welding of $1\,1/2$-In. Steam Pipe Lines with the Metal Arc. A series of $1\,1/2$-in. pipe sections, 12 in. long, were welded together. Both the closed and open butt joints were used. Two layers of weld metal were deposited, each one starting at the top of the joint, working downward on one side and up the other side, and completing the weld at the point of starting. The welded sections were tested with a hydraulic pressure of 100 lb. and the slight porosity found in a few joints was readily stopped by peening. Pressure was then raised to 2000 lb., without any more porosity being evident.

The closed butt joint is preferable to the open butt on pipes of small sizes because it eliminates "icicles" on the inside of the pipe and permits an intermingling of the base metal on both sides of the joint with the possibility of reducing porosity to a minimum.

The makes of electrodes used did not materially influence the porosity obtained in the weld. Porosity was usually found in the metal deposited in the overhead position; however, all porous spots can be and were readily stopped by peening.

The Application of Arc Welding to Building Construction, formed the subject of a paper by C. J. Holslag, Manager, Electric Arc Cutting and Welding Co. From what the author says it would appear that the use of welding may bring about the employment of entirely new shapes. He states, for example, that webs of angles, channels, and I-beams are not always necessary, and the strength in structure may be provided by other means. As an

instance of this he cites a truss practically made of rods welded together. The bottom and top chords as well as the struts are ingeniously formed out of bent rods arc-welded together. Where they meet at each end, they are joined by the same process to a standard structural shape. The truss is a real truss of the Pratt type, and when used with concrete or steel as a floor or ceiling, is said to produce a most economical construction. At Teaneck, N. J., a building was erected very similar to that of the wood-frame type, except that steel alone was used. It was built from flat and standard sections lifted into place and welded together.

Still another adaptation of arc welding lies in the attaching of roof purlins to the rafters, where a fireproof slab roof is used. Such a one is submitted by the U. S. Gypsum Co. or the Porete Mfg. Co. The standard T-bars, formerly drilled, bolted, or tapped in place, by hand, on the roof, are replaced by simply cutting and fitting the pieces and tacking them in place by arc welding. A single welder is able to attach in this way as much steel as a whole gang of iron workers. When it is realized that on gable roofs practically every piece has to be fitted in place on the job, the advantage of welding is obvious.

A list of buildings in the Metropolitan area that have been welded in this way is given, and New York is said to be not the only city to use this process.

Other papers not abstracted here were: Production Welding of Stern Frames by J. H. Deppeler, Metal and Thermit Corp.; Oxy-acetylene Welding of Furniture, by M. F. Bayer, Simmons Co.; Applications of Welding in Automobile Manufacture, by Robt. Appleton, Engr., Pierce-Arrow Manufacturing Co.; and a paper on Resistance Welding.

The Joint Pressure Vessel Committee of the A.S.M.E. and the A.B.W. met on the afternoon of April 29, with members of the American Welding Society participating.

The Administrative Board of A.E.C. on Flood Control

THE Administrative Board of American Engineering Council held a meeting in the Board Room of The American Society of Mechanical Engineers, New York City, on May 13, 1927. Among the actions taken were the following:

The proposal submitted by the American Society of Agricultural Engineers that the Division of Agricultural Engineering within the Bureau of Public Roads of the Department of Agriculture, be raised to the status of a bureau within the Department, was referred to the Public Affairs Committee for study and report.

Dr. William McClellan, Chairman of the Power Committee, submitted a tentative outline of a proposed meeting at which time the various economic phases of power development will be discussed. The Board approved the proposed program in principle and authorized the Committee to proceed with its plans, under the supervision of the Executive Committee. It is anticipated that the meeting will be held in New York City the latter part of October.

The Administrative Board gave consideration to the Bicentennial Anniversary of the Birth of George Washington. It was the sense of the Board that engineering groups throughout the United States should take an active interest and part in the celebration. The Executive Secretary was instructed to bring the matter to the attention of such groups and urge that they begin to lay plans for suitable participation of organized engineers in the celebration.

The Mississippi River situation was discussed and a resolution was passed calling upon the proper governmental authority to appoint at the earliest possible moment an engineering commission to examine into and report upon the broad problem of a comprehensive plan for control of the Mississippi River and the prevention of a repetition of the disastrous conditions now existing in consequence of the recent floods. The Commission is to consist of seven engineers, not less than five of whom shall be civilian engineers of national reputation in hydraulics or of experience and training especially qualifying them to examine into and report upon the problem of the Mississippi River, and who are not in the employ of the Corps of Engineers. The resolution further provided that the Ransdell Hydraulic Laboratory Bill be reintroduced into Congress and pressed to early passage, with ample appropriations to permit of immediate experimental investigations of flood conditions and effects.

Pursuant to instructions received some months ago, the Executive Secretary reported upon the preliminary survey of the status of legislation relating to stream pollution. The returns indicated that there were a number of groups who felt a national committee composed of representatives of all interested organizations should be formed for the purpose of coördinating the work being done with reference to stream pollution. In view of this the Administrative Board authorized the appointment of a committee to give consideration to the formation of such a national committee and to report at the next meeting of the Board.

Announcement was made that the report on Accidents and Production was entering upon the last stages of editing, with the anticipation that the manuscript would be placed in the hands of the printer within ten days.

The President announced the appointment of Messrs. H. E. Howe, Gardner S. Williams, and L. W. Wallace on a Committee on State Engineering Councils, which committee is to make a thorough study of the question of State Engineering Councils and report to the next meeting of the Administrative Board.

American Philosophical Society Bicentenary

THE Bicentenary Celebration of The American Philosophical Society was held in Philadelphia for four days, April 27 through 30, 1927. This pioneer American scientific society founded by Benjamin Franklin was an outstanding factor in stimulating scientific thought in the early days of the republic. It is the forerunner of the many powerful learned societies in the country today. It is fitting that the celebration of two centuries of useful service should be marked by a comprehensive program of addresses and papers by leaders in all fields of science. It would be interesting to speculate on the wonder in the minds of the founder of this venerable society if they were to sit now in a gathering to hear learned presentations of the endocrine organs, acoustic impedance of air columns, spectroscopy, the atomic number of uranium, the first description of the North Pole, photoelectric photo-engraving, lipoids in plant cells, and gaseous metabolism in wild birds.

A cloud was thrown over the meeting by the death of the society's president, Dr. Charles Walcott, just previous to the dates set for it, Vice-Presidents Henry F. Osborn, W. W. Campbell, and Francis X. Dercum officiated at the sessions. The American Society of Mechanical Engineers was represented by Past-President Ambrose Swasey, William Elmer, and Conrad N. Lauer, Members of Council, and Secretary Calvin W. Rice.

Economic Production of Steam by Electricity

THE author makes a general statement that where coal is 30 shillings ($7.50) per ton and electricity 0.1 d. (0.2 cent) per unit, coal might economically be displaced by electricity for the production of such steam as is required for purposes other than power, each case, however, requiring individual investigations. In the plan described here several thousand kilowatts of energy are taken continuously and electricity exclusively is used for power. For process work, however, in the manufacture of one product there is required a quantity of steam amounting to 20,000 lb. per day and this amount is likely to be doubled shortly. The steam is required at a pressure between 10 and 20 lb. and in the past a range of fired boilers were used to produce it.

In a preliminary investigation the author found that maintenance costs on electric boilers were very low, in one case an assertion by the City Engineer of Freiburg having been made that in the course of two years no costs had to be debited to the electric boiler for maintenance. The space occupied by the electric boilers is very small, in one case an electric boiler occupying with switch gear not more than 7 ft. × 7 ft., doing the same work as three Lancashire type boilers. The author has not yet installed an electric boiler but merely made a preliminary investigation. (C. J. Wharton in a paper read before the Institution of Engineering Inspection, London, Feb. 25, 1927, abstracted through *Combustion*, vol. 16, no. 5, May, 1927, pp. 271–273.)

FIG. 1 MACHINE-TOOL EXHIBITION HALL ON THE GROUNDS OF THE TECHNICAL EXPOSITION

The German Machine-Tool Exhibit at Leipzig

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THE machine-tool exhibit at the Technical Fair in Leipzig is of interest to American engineers and tool builders in view of the machine-tool exhibits in New Haven and the forthcoming exhibit of the National Machine Tool Builders' Association at Cleveland.

The German exhibit is part of the Technical Fair held for two weeks during March in connection with the general Leipzig Sample Fair. The Leipzig Fair is the largest and oldest fair now in existence. Its beginnings are lost in the Middle Ages. It was well established by the 12th century and put on a firm basis in the 15th century, when the German Emperor, Frederick III, recognized it and insured safe conduct to travelers and goods to and from Leipzig. By 1768 it was visited by more than 8000 merchants. By 1839 this number was increased to over 38,000 and today upward of 150,000 visit it. In the early days, the goods themselves were brought to Leipzig and sold by wholesalers to retailers. Gradually it became a Sample Fair, and today its chief function is to facilitate business between wholesalers and manufacturers. The general Sample Fair is held twice a year, but the Technical Fair is held once a year only, for two weeks during the month of March.

The Machine-Tool Exhibition is housed in a splendid building known as Hall No. 9, shown in Fig. 1, built a few years ago at a cost of approximately five million marks. It is of permanent steel and concrete construction with three large steel arched bays in the center and a smaller bay with a long gallery along each side. It is remarkably well lighted, as will be seen from Fig. 2. The ground-floor exhibition space is approximately 250 by 550 ft.; and the additional space in the two side galleries, combined, is nearly 100 by 550 ft. The building is supplied with electric current a.c. and d.c., and each of the three large bays has crane service for handling exhibits. In it are housed the exhibits of the members of the five Vereins covering the manufacturers of metal- and woodworking machinery, precision tools, grinding mediums, and punches and dies. The membership of these Vereins is about 350. Of these 216 are exhibiting this year. Last year there were 208.

In another building known as Hall No. 11 are, among other exhibitors, the foreign tool manufacturers and certain machine-tool dealers importing foreign machinery. Those in Hall 11 showing machine tools number about 40.

The area occupied by the Technical Fair, of which the machine exhibit is a part, comprises a group of buildings in the southern part of the city, providing space for 20 or more groups of exhibits in addition to machine tools, covering such fields as power and electrical machinery, materials handling and transportation, hardware, heating and ventilation, etc. The total number of exhibitors in the technical section in 1925 was 2494; in 1926, 2261. This year it is somewhat higher. These figures show that the Technical Fair exhibitors average about 25 per cent of the exhibitors in the whole Leipzig Sample Fair.

From the technical point of view the tool exhibits show the skill of the German mechanics and engineers and deserve careful attention on the part of American engineers and tool builders. It has been charged that the German tool builders have been merely copying American machinery, and it is true that in this year's fair there are machines bearing German names which look very familiar to an American. But the engineering skill of Germany is quite capable of going forward independently, and there are refinements and new developments shown which deserve careful attention from those who keep abreast of the best practice. Reuleaux forecast this condition in the following statement which is quoted in the preface of the official guide to the machine-tool division of the Leipzig Fair of 1927:

"The amazing development which the art of machine construction has experienced did not begin until the advent of that peculiar

[1] EDITOR'S NOTE.—Joseph W. Roe, the author of this article, is head of the Department of Industrial Engineering at New York University and is a keen observer of the development of production methods and production machinery. He has recently returned from an extended European trip, during which he visited many of the industrial centers, the great industrial museums, and the Leipzig Fair. He was deeply impressed by the technical exhibits at Leipzig this spring, those of machine tools in particular, and this statement of his impressions should be of keen interest to American engineers and manufacturers in this day of growing international trade in industrial machinery.

and correct change in the conception of engineering inventors which has brought to pass that machines are no longer intended to imitate the work of hands or even to imitate Nature at all, but are designed to solve each task with their own characteristic and appropriate means, differing perhaps very widely from the means employed by Nature."

The exhibition is so large that a detailed description of it is not possible here. In general, it may be said that all but a small portion of the machines are electrically driven. Some of them have variable-speed reversing hydraulic drives, giving an infinite speed control between the limits set. Some of them appear to be more complicated than American shop practice would justify, but it is claimed that they are working very satisfactorily under production conditions, and their smooth operation and ease of control under demonstration certainly were remarkable.

The exhibit of sanding machines for woodworking, of all types, showed great activity in this field. Zeiss of Jena had a fine exhibit of precision gages and of methods of applying optical measurements, based on the wave length of light, to machine measurements. There were quite a number of automatic lathes for high-speed work, of course operating on the cam principle, but doing away with the turret, the tools being brought into operation by fingers swinging into place for the cutting position. The "idle time" of these automatics seems to have been reduced to an absolute minimum, and this fact, coupled with the remarkably high spindle speeds and tool feeds made possible the production of small pieces in what is described as "split-second" time.

Schiess-Defries had a remarkable horizontal boring machine for machining such parts as large turbine casings. It was far and away the largest tool of this type which the writer has ever seen. Another large machine was one for the complete assembling and final grinding of the largest-sized armatures. It was electrically operated, with very exact rotational control, down to small parts of a revolution.

As might be expected, there was a strong exhibit of gear-cutting machines, especially of the hobbing type, among these being a new machine for milling bevel gears with a tooth form somewhat between the herringbone and crown type.

Every detail of the exhibition was very carefully thought out and there were no evidences of haste or carelessness in preparation. The machines were well shown in every case and the printed guide, by a very simple and easily understood arrangement, enabled any person speaking any one of five languages—English, Spanish, Italian, French, or German—to at once locate the exhibit of any particular machine in which he might be especially interested.

In its broader aspects this exhibit is an example of the new selling methods which are spreading throughout Europe since the war. Not only has the Leipzig Fair broadened rapidly and increased in importance, but there are now sample fairs held annually or oftener at Basle, Prague, Lyons, Birmingham, Milan, and Vienna. Practically all of these, with the exception of the Leipzig Fair, have sprung up since the War. The first and main objective in each case is to develop export trade by supplying a time and place where foreign buyers can see samples of the export products of the country. There is a growing movement to encourage exhibits by foreign countries of products which can be imported to advantage, the idea being that exchange of goods leads to more active trade and advantageous commerce than export trade alone. American engineers and manufacturers should watch this development closely. This applies to machine tools as well as to the general articles of commerce. The management of the Leipzig Technical Fair state that they welcome not only American visitors but American exhibitors.

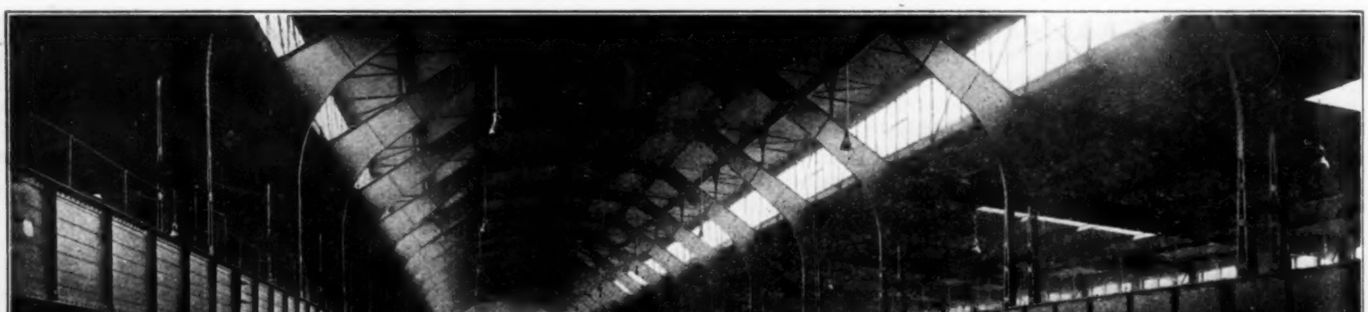

Fig. 2 General View of the Machine-Tool Exhibit

Book Reviews and Library Notes

The Library is a coöperative activity of the A.S.C.E., the A.I.M.E., the A.S.M.E. and the A.I.E.E. It is administered by the United Engineering Society as a public reference library of engineering and the allied sciences. It contains 150,000 volumes and pamphlets and receives currently most of the important periodicals in its field. It is housed in the Engineering Societies Building, 29 West 39th St., New York, N. Y. In order to place its resources at the disposal of those unable to visit it in person, the Library is prepared to furnish lists of references on engineering subjects, copies of translations of articles, and similar assistance. Charges sufficient to cover the cost of this work are made.

The Library maintains a collection of modern technical books which may be rented by members residing in North America. A rental of five cents a day, plus transportation, is charged. In asking for information, letters should be made as definite as possible, so that the investigator may understand clearly what is desired.

Books Received in the Library

Browns' Directory of American Gas Companies and Gas Engineering and Appliance Catalogue. 1926 edition. Robbins Publishing Co., New York, 1926. Cloth, 9 × 12 in., 831 pp., illus., $10.

This annual compendium is a convenient source of information upon many questions that arise frequently in the gas industry. The catalog section is a quick guide to the firms that deal in gas-works apparatus and supplies and to the books treating of the gas industry. The directory section gives the more important statistical data about the companies manufacturing and supplying gas in the United States, with name and place indexes. It also lists the public-service commissions and gas associations and the members of the latter.

Cotton: History, Species, Varieties...Culture...Marketing and Uses. By Harry Bates Brown. McGraw-Hill Book Co., New York, 1927. Cloth, 6 × 9 in., 517 pp., illus., diagrams, tables, $5.

Intended as a compact survey of the cotton industry in its various aspects. The author who is professor of cotton breeding at Louisiana State University, covers cotton growing from the selection of the seed to the marketing of the cotton and the seed.

Diesel Engines, Marine and Stationary. By Arthur H. Goldingham. Third edition. E. & F. N. Spon, London, 1927. Cloth, 6 × 9 in., 255 pp., illus., diagrams, plates, 21s.

The great progress in the design and construction of Diesel engines, which has taken place since the second edition of this work appeared, has led to rewriting of the greater part of the book and to thorough revision throughout. The work is wide in scope, giving attention both to design and to operation. It includes many drawings and illustrations of late types of engines, and gives data on the leading varieties of the Diesel engines of various makers.

Interpolation. By J. F. Steffensen. Williams & Wilkins Co., Baltimore, 1927. Cloth, 6 × 9 in., 248 pp., $8.

This work owes its existence to the lectures given by the author in recent years to actuarial students at the University of Copenhagen, and is, except for minor emendations, a translation of the Danish edition published in 1925. It is intended as a text-book and not as a treatise or handbook. The point of view adopted is that only those approximate formulas are to be included for which it is possible to derive a remainder-term simple enough to permit the calculation of limits to the error involved in the formula, and the aim has been to meet the needs of the practical computer rather than those of the mathematician.

Lectures on Theoretical Physics, vol. 1. By H. A. Lorentz. Macmillan Co., London and New York, 1927. Cloth, 6 × 9 in., 195 pp., $4.

The first of a series of volumes which will present the courses of lectures which Professor Lorentz has delivered at the University of Leyden. Two of these courses, one upon Ether Theories and Ether Models, delivered in 1901–1902, and one upon Kinetical Problems, delivered in 1911–1912, are contained in this first volume.

The first course describes the attempts of Stokes, Maxwell, Kelvin, and others to account for various phenomena, especially electromagnetic ones, by means of speculations about the structure and properties of the ether. The second discusses some questions belonging partly to the domain of the kinetic theory of gases and partly to that of the electron theory.

This translation of the lectures of the eminent physicist will be welcomed by all students of theoretical physics.

Marvels of Modern Mechanics. By Harold T. Wilkins. E. P. Dutton & Co., New York, 1927. Cloth, 6 × 9 in., 280 pp., illus., $3.

Contents: Conquest of the atom; strong rooms of the sea bed; probing the sun's secrets; far north and the seal hunters; wonders of wires and wireless; marvels of modern excavation; charting unknown seas today; human flashlights underground and a contrast; making the earth work; cable ship on the high seas; safer airways of today; scenic art of the modern theater; revolution on the ocean.

A popular account of recent discoveries and inventions, intended for the general reader. Calls attention to many recent advances in engineering, physics, and chemistry.

Metallographic Researches. By Carl Benedicks. McGraw-Hill Book Co., New York, 1926. Cloth, 6 × 9 in., 307 pp., illus., diagrams, $4.

Contents: Some points of view on the kinetic constitution of solid matter; homogeneous thermoelectric and electrothermal effects; hardness in general and the hardening of carbon steel; high-speed steel and other alloys; meteoric iron and invar; some improvements in high-power microscopy; microchemical etching method for metallography; first discoverer of the critical point Ac in steel; some fundamental factors for obtaining sharp thermal curves; determination of the specific gravity of molten iron; rational ingot section for hard-rolled material; action of hot wall a factor of fundamental influence on the rapid corrosion of water tubes and related to the segregation in hot metals; index.

Part of the contents of this volume is made up of lectures delivered before the Division of Metals of the American Institute of Mining and Metallurgical Engineers and other organizations. As the table of contents indicates, the text discusses a somewhat heterogeneous collection of topics which interest the metallographist and metallurgist, the subjects being those which have recently been investigated at the Institute of Metallography, Stockholm, under the direction of the author.

The Oil War. By Anton Mohr. Harcourt, Brace & Co., New York, 1926. Cloth, 5 × 8 in., 267 pp., $2.50.

The purpose of this book is to describe the political struggle for control of the oil supply of the world. Starting with an account of the origin and organization of the great American, Dutch, British, and Russian companies, the author discusses the oil policies of the various governments, the part played by oil in the Great War, the situation since its ending, and the probable course of developments. Mr. Mohr, it happens, is a citizen of a state that is not affected directly by the diplomatic aspect of the question, and his story is a graphic and interesting description of the situation as it exists today.

Practical Radio Construction and Repairing. By James A. Moyer and John F. Wostrel. McGraw-Hill Book Co., New York, 1927. Cloth, 5 × 8 in., 319 pp., diagrams, tables, $2.

Intended as a guide for amateur constructors of receiving sets and for those who wish to know how to make minor repairs and improvements. Directions are given for building, testing, and repairing of the important types of sets and such commonly used equipment as wave traps, trickle chargers, battery eliminators, loud speakers, etc. Superheterodyne, short-wave, impedance-coupled, and resistance-coupled sets are given special attention.

Problems of Modern Physics. By H. A. Lorentz. Ginn & Co., Boston and New York, 1927. Cloth, 6 × 8 in., 312 pp., 8 × 6 in., $3.60.

These lectures are a scholarly discussion of many important questions confronting the physicist of today by one of the leading workers in that science. The propagation of light in different media, the structure and mutual action of atoms, the ideas of the theory of quanta and their application to the Zeeman effect, and the theory of relativity and its effect on the fundamental laws of dynamics, are among the subjects.

Processes of Flour Manufacture. By Percy A. Amos; new edition revised by James Grant. Longmans, Green & Co., London and New York, 1925. (Longman's technical handicraft series.) Cloth, 5 × 8 in., 311 pp., illus., diagrams, tables, $3.

A thorough yet concise textbook on milling, based on a course of instruction for those actively engaged in the industry, given at the Manchester College of Technology. Although primarily adapted to British conditions and practice, the book will be of interest to American millers and is a welcome addition to the scanty literature on the subject.

Properties and Testing of Magnetic Materials. By Thomas Spooner. McGraw-Hill Book Co., New York, 1927. Cloth, 6 × 9 in., 385 pp., illus., diagrams, $5.

In this book the principal objects are to give a reasonably complete summary of the magnetic properties of commercial ferromagnetic materials, to describe the useful methods of testing for commercial inspection and scientific investigation, and to examine critically the accuracy and limitations of these methods. The work is intended for those who wish definite information for use in the solution of some specific magnetic problem, and therefore does not discuss matters of theory or topics that have no engineering significance. The choice of data and information is based upon the lengthy experience of the author with the problems that arise in the design and construction of electrical apparatus.

Radio Encyclopedia. Compiled by Sidney Gernsback. Sidney Gernsback, New York, 1927. Leatherette, 9 × 12 in., 168 pp., illus., diagrams, $2.

A convenient compilation of radio terms and data, arranged alphabetically and accompanied by a classified index. Nearly two thousand words and terms are defined and many diagrams of circuits and apparatus are included. The book is a handy work of reference.

Raschlaufende Ölmaschinen. By Otto Kehrer. R. Oldenbourg, Munich and Berlin, 1927. Paper, 8 × 11 in., 111 pp., diagrams, plates, tables, 10 r. m.

Presents the results of a thorough experimental and theoretical study of a modern hot-bulb high-speed oil engine, discusses the behavior of high-speed Diesel engines of various designs, and concludes with a comparison of these with gas engines. The investigation was made at the Munich Technical High School.

Verwertung von Abfall- und Überschussenergie. By E. H. de Grahl. V.D.I. Verlag, Berlin, 1927. Cloth, 9 × 12 in., 305 pp., illus., diagrams, plates, map, tables, 22 r. m.

Dr. de Grahl's theme is the conservation of fuel through the utilization of energy that is usually wasted. His book is divided into two parts. The first treats of the recovery of fuel from the ashes of power plants, coke works, and other establishments, and of methods for utilizing it by briquetting, gasifying, low-temperature carbonization, etc. It also discusses the utilization of the excess power generated at coke works, blast-furnaces, gas works, and other industrial plants.

Part two discusses methods for economizing steam by improving steam engines and installing accumulators and preheaters and by district heating. The book is a useful review of present practice.

S.A.E. Handbook, March, 1927. Society of Automotive Engineers. Published by the Society, New York, 1927. Fabrikoid, 5 × 8 in., various paging, illus., diagrams, tables, $5.

The Handbook contains the standards and recommended practices of the Society of Automotive Engineers, including the modifications adopted since the previous publication in October, 1926. Sixteen revisions are recorded, four new standards are given, and eight former practices are cancelled.

The recommendations cover the materials entering into the construction of internal-combustion engines, automobiles, motor boats, and airplanes. Many of them have, of course, much wider applicability than within the field of the Society, and the book is in consequence a useful reference work to builders of machinery generally.

Treatise on the Mathematical Theory of Elasticity. By A. E. H. Love. Fourth edition. Cambridge University Press, London; Macmillan Co., New York, 1927. Cloth, 7 × 11 in., 643 pp., $13.50.

The fourth edition of this work, the classical treatise in the English language, will be welcomed by mathematicians; for while it does not present any marked changes from the second and third editions, it is has been revised and extended to cover the researches published since 1914.

The most important additions are concerned with the theory of a rectangular plate, clamped at the edges and bent by pressure applied to one face; the theory of the resistance of a very thin plate to pressure; and the process by which stress-strain relations are deduced from the molecular theory of a crystalline solid. The theory of the equilibrium of a sphere has been simplified and its application to geophysical problems made easier.

In addition to being a masterly exposition of the subject, the book is noteworthy for its very complete set of references to the original papers and memoirs dealing with the theory of elasticity.

Treatise on Thermodynamics. By Max Planck. Translated by Alexander Ogg. Third edition from the seventh German edition. Longmans, Green & Co., London and New York, 1927. Cloth, 6 × 9 in., 297 pp., $5.

This edition has been carefully revised, corrected, and extended to bring it into agreement with the latest German edition. It is intended as an introductory textbook for students who have an elementary knowledge of physics, chemistry, and the calculus.

Über die Geschichtliche Entwicklung der Wollkämmaschine und ihre Technologische Arbeitsweiss. By Hans Richard Wolf. V.D.I. Verlag, Berlin, 1927. Paper, 6 × 8 in., 109 pp., illus., diagrams, 4.80 mk.

Since the publication of Lehren's work on carding machines, in 1875, no one has attempted to write a history of these complicated, ingenious devices. Dr. Wolf's book will be welcome to designers of textile machinery as well as to others interested in the construction of kinematically complicated machines, for it describes the latest developments in carding and illustrates fully the various mechanisms that have proved of lasting value.

Starting with Cartwright's patent of 1789, the book traces the systems that have been tried and describes the machines belonging to each. The genealogy of modern patterns is carefully brought out.

Wärmestrahlung Technischer Oberflächen bei Gewoehnlicher Temperature. By Ernest Schmidt. R. Oldenbourg, Munich and Berlin, 1927. (Beihefte zum *Gesundheits-Ingenieur*, Reihe 1, heft 20.) Paper, 9 × 12 in., 23 pp., diagrams, tables, 3.60 mk.

An investigation of radiation from structural materials heated to temperatures between 70 and 95 deg. fahr., special attention being paid to the effect of surface conditions. The materials tested included rough and polished brass; tinned, galvanized, and nickel-plated iron; polished and oxidized copper; rough and polished aluminum; rough iron and steel; painted and varnished surfaces; wood; rubber; brick; glass; roofing paper; marble; etc. A brief review of the laws of radiation, so far as they have technical importance, is also given.